

Experimental Study of the Effect of Air Filter Pressure Drop on Internal Combustion Engine Performance

by
Tadeusz Dziubak



* and
Mirosław Karczewski



Faculty of Mechanical Engineering, Military University of Technology, 2 Gen. Sylwestra Kaliskiego Street, 00-908 Warsaw, Poland

Author to whom correspondence should be addressed.

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Abstract

The paper presents the problem of the effect of air filter pressure drop on the operating parameters of a modern internal combustion engine with compression ignition. A literature analysis of the results of investigations of the effect of air filter pressure drop on the filling, power and fuel consumption of carburetor and diesel engines with classical injection system was carried out. It was shown that each increase in the air filter pressure drop Δp_f by 1 kPa results in an average decrease in engine power by SI 1–1.5% and an increase in specific fuel consumption by about 0.7. For compression ignition engines, the values are 0.4–0.6% decrease in power and 0.3–0.5% increase in specific fuel consumption. The values of the permissible resistance of the air filter flow $\Delta p_{f\text{dop}}$ determined from the condition of 3% decrease in engine power are given, which are at the level of 2.5–4.0 kPa—passenger car engines, 4–7 kPa—truck engines and 9–12 kPa—special purpose vehicles. Possibilities of decreasing the pressure drop of the inlet system, which result in the increase of the engine filling and power, were analyzed. The program and conditions of dynamometer engine tests were worked out in respect to the influence of the air filter pressure drop on the operation parameters of the six-cylinder engine of the swept volume $V_{ss} = 15.8 \text{ dm}^3$ and power rating of 226 kW. Three technical states of the air filter were modeled by increasing the pressure drop of the filter element.

For each technical state of the air filter, measurements and calculations of engine operating parameters, including power, hourly and specific fuel consumption, boost pressure and temperature, were carried out in the speed range $n = 1000\text{--}2100$ rpm. It was shown that the increase in air filter pressure drop causes a decrease in power (9.31%), hourly fuel consumption (7.87%), exhaust temperature (5.1%) and boost pressure (3.11%). At the same time, there is an increase in specific fuel consumption (2.52%) and the smoke of exhaust gases, which does not exceed the permissible values resulting from the technical conditions for admission of vehicles to traffic.

Keywords:

internal combustion engines; air filter pressure drop; engine power; hourly and specific fuel consumption; smoke opacity

1. Introduction

The basic task of the intake system of an internal combustion engine of a motor vehicle is to supply air from the environment to the engine cylinders in appropriate quantities and with appropriate pressure, temperature and density, so as to ensure the correct course of the fuel combustion process in the cylinders and optimize the engine performance [1,2,3,4]. An engine requires at least 14.5 kg of air to burn 1 kg of fuel. When operating at rated conditions, passenger car engines draw 150–400 m³/h of air per hour. For truck engines this value is 900–2000 m³/h, and for a special vehicle engine e.g., Leopard 2 tank—more than 6000 m³/h. The air is added to the combustion process in order to obtain the required engine performance.

Various pollutants—gaseous and solid—get into internal combustion engines together with the air. The basic component of mechanical impurities is mineral dust, which is lifted from the ground to the height of several meters by the movement of vehicles or the wind, and then sucked in by the intake air intake [5,6]. The basic components of mineral dust are SiO₂ and Al₂O₃ grains, which are characterized by sharp edges, very high hardness and account for 65–95% of the total dust mass depending on the type of substrate atmospheric conditions [7,8].

In internal combustion engines, mineral dust grains are carried with the air into the piston crown where they settle onto the surface of the cylinder liner and piston. Together with the oil, the dust forms an abrasive paste which, when it comes into contact with the surfaces of the engine components P-PR-C (piston–piston ring–cylinder), causes abrasive wear. As a result of excessive wear of the “piston–cylinder” pair, there is a decrease in the leak-tightness of the piston crown and the resulting drop in charge pressure at the end of the compression stroke, and consequently a decrease in the engine’s compression ratio and power as well as an increased blow-by of hot combustion gases into the oil sump, which cause an increase in the temperature of the P-PR-C bonding elements and the oil. This leads to a decrease in oil viscosity and to excessive wear of cylinder rings and cylinder liner [9]. To counteract this phenomenon, special composite (diamond-like and titanium) coatings are applied to the piston ring surfaces [10,11,12].

From the point of view of abrasive wear, the most dangerous are dust particles, whose diameter d_p is equal to the thickness of the oil film h_{min} between two surfaces at a given moment. In typical combustion engine associations, the oil film thickness h_{min} does not exceed the value of 50 μm [9,12].

In order to minimize the wear of engine components, an air filter is installed in the intake system, which ensures that the air supplied to the engine cylinders is of suitable quality (purity) in terms of mechanical components. Air filters differ in principle and method of operation, construction, type of filtration baffle and efficiency of operation. The passenger car engines, which are operated at low concentrations of air dust, are equipped with single-stage (baffle) filters, where the filtering element is a commonly used cartridge (rectangular panel), made of pleated filter paper or non-woven filter cloth [13,14,15]. Engines of trucks, special vehicles (tanks, armored personnel carriers, infantry fighting vehicles) and working machines operating in conditions of high dust concentrations in the air are usually equipped with two-stage filters. The first stage of air filtration is a multicyclone and the second one is a cylindrical filtering insert which, due to space limitations and efforts to increase the surface area of the filtering material, is made of pleated filter paper [16,17].

The other tasks of the air intake system are damping of the noise caused by the air flow [18] and forcing the wave phenomena, causing so-called dynamic (resonant) supercharging in the desired ranges of engine work, and thus increasing the cylinder filling and engine power [19]. The other elements of the air supply system of the internal combustion engine which are involved in the supply of air to the engine

cylinders are: the flow meter, the turbocharger, the charge air cooler, the air damper, the intake manifold, the intake ducts in the head and the intake valves. The flow meter, located downstream from the air filter, continuously determines the mass of air flowing and transmits this information to the ECU (Engine Control Unit).

The purpose of the turbocharger (standard equipment in diesel engines) is to increase the mass of working medium supplied to the engine's cylinders by increasing the pressure (and temperature) of the air sucked into the system. The air cooler lowers the temperature of the charge air (its density increases) and thus increases the mass flow of the air supplied to the engine cylinders. In order to measure the permissible resistance of the air filters of trucks and special-purpose vehicles, sensors are used, which are mounted on the air filter outlet duct. The crankcase ventilation system and the EGR (exhaust gas recirculation) system are related to the air supply system.

A characteristic feature of baffle filters is that, during operation, as a result of deposition and accumulation of dust particles in the filter bed, the filter pressure drop Δp_f defined as the static pressure drop behind the filter increases systematically (Figure 1). The intensity of the increase of the pressure drop depends on the conditions in which the vehicle is operated, and mainly on the dust concentration in the air and the engine operating time. The higher the value of the dust concentration in the air sucked into the engine, the faster the filter reaches the permissible value— $\Delta p_{f\text{dop}}$. The dust retention process is strongly dependent on the size of the incoming particles [20,21]. As the number of fine particles arriving with the air on the filter bed increases, which are characterized by a larger surface area per unit mass than particles of large size, a layer of difficult permeability forms on the filter layer. Therefore, the pressure drop of the filtration bed increases rapidly and the filter service life, limited by the $\Delta p_{f\text{dop}}$ value, becomes shorter [22,23].

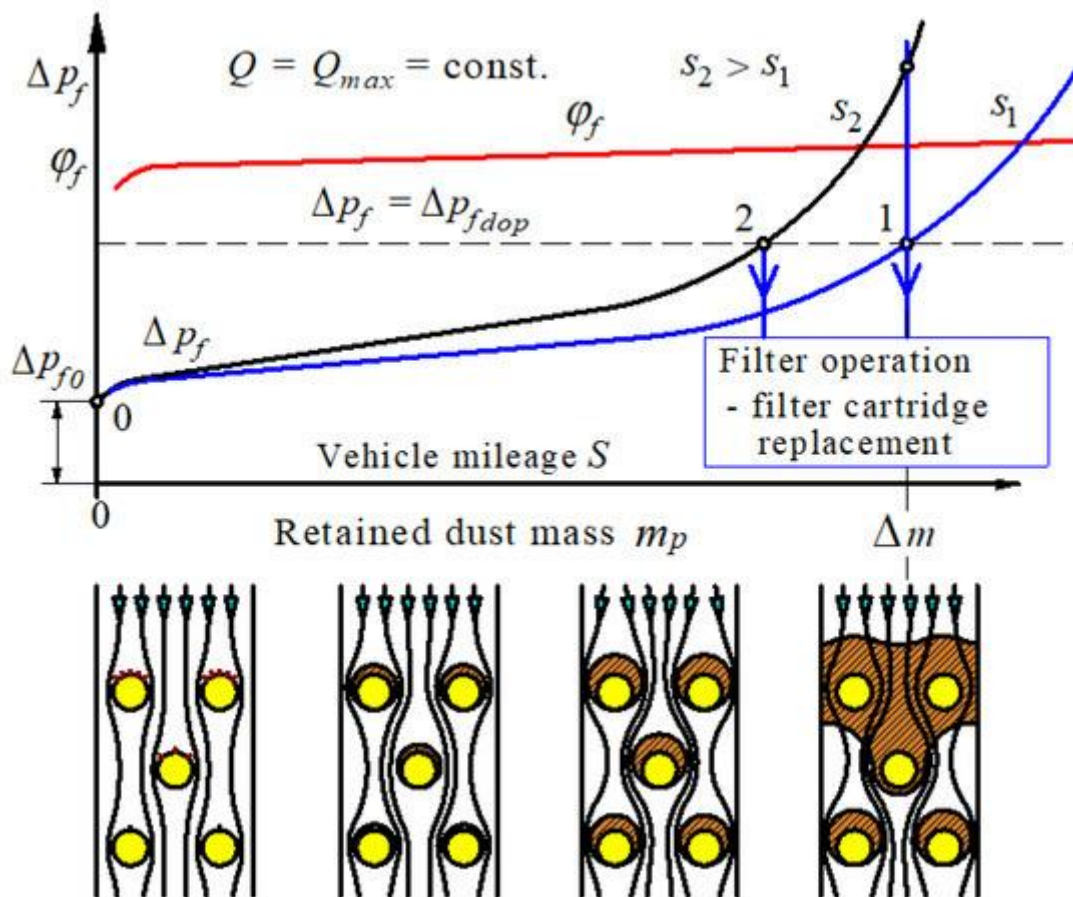


Figure 1. Changes in filtration efficiency and pressure drop of baffled air filter during operation: 1—low air dustiness, 2—high air dustiness.

The filtration process is also strongly dependent on the type of pollutants. Soot coming mainly from engine exhaust emissions very quickly forms a tight, greasy layer on the surface of the fibrous filter bed, which results in a more intensive increase in pressure drop and a faster attainment of the Δp_{fdop} value [24].

When the dust absorption capacity of the filter bed is exhausted (Point 1 in Figure 1), there is a sharp increase in the pressure drop of the air filter. The filter pressure drop Δp_f corresponding to the pf value is called the permissivity resistance Δp_{fdop} .

As dust settles inside or on the surface of the filter material, the air flow rate tends to decrease due to increasing resistance to air flow through the filtration system. Operating an engine with an air filter with increased pressure drop causes a decrease in fill and torque, resulting in a decrease in maximum engine power. The result is a reduction in the dynamic properties of the vehicle. The high resistance of the air filter and the significant negative pressure created behind the filter can cause the forces of detachment to exceed the forces of adhesion of the grains to the filter substrate, resulting in the dust being detached in an avalanche (so-called secondary emission) and being sucked into the engine cylinders along with the air, causing accelerated wear of the P-PR-C elements. In extreme cases, a high pressure drop may cause mechanical damage (rupture, cracking) of the filter insert (Figure 2), which will result in intensive wear of the engine elements.

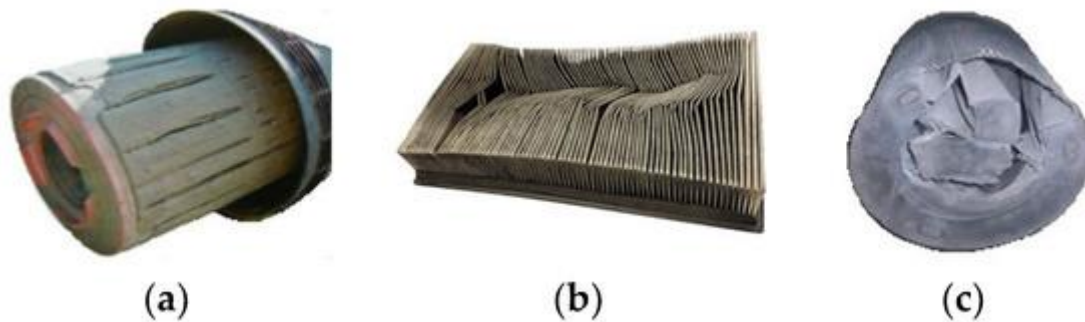


Figure 2. Effects of operating an air filter beyond the permissible resistance: (a) excessively contaminated filter element, (b,c) filter element failure.

For this reason, when the filter reaches a certain resistance value, it is necessary to service the air filter by replacing the filter cartridge. In passenger cars, due to low values of dust concentration in the air and thus small increments of the filter pressure drop, this operation is performed depending on the mileage of the vehicle or its operation time. These values vary considerably, depending on the design specifics of the engines and the car manufacturer, ranging from 15,000–90,000 km or from one to 4 years [25,26].

According to the authors of [27,28], in the case of passenger cars, the air filter should be replaced when the increase in the pressure drop reaches 1.0–2.5 kPa. The research presented in [29,30] has shown that the replacement of the filter insert, depending on the mileage of the vehicle or its operation time, may be premature—before the full dust absorption capacity and the permissible pressure drop are reached, or too late. In this case, the increased filter pressure drop will cause a decrease in engine filling and deterioration of engine characteristics—a decrease in power output.

For trucks and special vehicles, which are operated in conditions of high and variable concentrations of dust in the air, thus, the increase of the pressure drop is significant (by 5–8 kPa) and occurs with different intensities. Failure to replace the filter cartridge on a regular basis, prolonging the mileage of the vehicle, results in an additional increase in the pressure drop and, as a consequence, a significant drop in the engine power and reduction of the vehicle's dynamics. At the same time, the filter still provides high efficiency and filtration accuracy.

For this reason, after the air filter of the engines of vehicles operating in conditions of high dust concentration in the air, sensors are installed, signaling the achievement of the set value of the permissible pressure drop Δp_{fdop} , which is a signal to operate the air filter. The value of the permissible resistance Δp_{fdop} is determined from the condition of a 3% decrease in engine power and is 2.5–4.0 kPa—passenger car engines—4–7 kPa—truck engines [31]—and 9–12 kPa—special purpose vehicles [32].

Technically, air filter life is commonly defined as the level of restriction that causes the pressure on a passenger car filter to drop by about 2.5 kPa above the pressure drop of a new (clean) filter. For trucks and special vehicles, Δp_{fdop} values are assumed to be about 6.25–7.5 kPa above the pressure drop of a clean air filter [33]. Therefore, efforts are made to minimize the pressure drop of clean air filters to reduce engine

energy loss and extend vehicle mileage. Air filter performance is a technical trade-off between pressure drop (vehicle mileage), filtration efficiency and accuracy, and engine wear and durability and vehicle reliability [34,35].

In light of the comments made, it can be thought that:

(1)

There are no unambiguously defined values of the engine power drop resulting from the need to overcome the air pressure drop. Thus, it is impossible to determine the resulting permissible values of the air filter pressure drop.

(2)

The literature does not give unequivocally the nature of changes in the engine performance—decrease in useful power, increase in smoke, increase in specific fuel consumption and smoke of the engine exhaust, depending on the air pressure drop.

In the available literature, it is difficult to find a sufficiently complete picture of the influence of the air pressure drop on the engine operation indicators, and in particular on its performance characteristics. Such an assessment may be obtained during experimental tests of the engine on an engine dynamometer or of the whole vehicle on a chassis dynamometer. However, these tests are costly and labour-intensive, which explains the limited number of results available. Nevertheless, it is the most reliable research method. In the available literature, there are few experimental results available on the effect of the pressure drop on the performance parameters of naturally aspirated carburetor and diesel engines with classical injection system—with an in-line, sectional injection pump.

Electronically controlled direct petrol injection systems are commonly used to prepare the mixture in modern high speed spark ignition internal combustion engines. They enable precise control of the fuel dose and the beginning of fuel injection, which, together with the information from the air flow meter, makes it possible to determine the optimum fuel mixture composition for the operating conditions, and, as a result, reduce the emission of toxic exhaust components. Passenger cars are mostly equipped with spark-ignition engines with direct injection of petrol and electronic control system, while truck engines are equipped with high-pressure diesel injection systems such as common rail, electronically controlled pump-injector systems. The engine ECU controls all the quantities that affect the value of the generated torque produced by the engine, while at the same time meeting the requirements in the area of exhaust emissions and fuel consumption throughout the life of the vehicle.

In a modern passenger vehicle engine air supply system, there is a flow meter, most often an HFM layer thermoanemometer, which measures the mass of air drawn in by the engine and transmits the appropriate signal to the engine control system. In the exhaust system, on the other hand, there is a λ probe, which continuously monitors the amount of oxygen in the exhaust gases, and its signal is used to optimize the composition of the mixture with regard to the content of toxic compounds in the exhaust gases.

Increasing resistance of the air filter during operation decreases the value of the mass flow of air, which may cause disturbances in the process of mixture preparation and combustion. In the available literature, there are no results of investigations of the influence of the air pressure drop on the work of modern SI engine (spark ignition engine) as well as compression ignition engine. In this paper it was decided to partly fill this gap by carrying out experimental tests on an engine dynamometer on the influence of the air pressure drop on the work parameters of a modern compression ignition engine used to drive a truck tractor. The basic operating parameters of the engine were determined as a function of the engine speed, including: power, hourly and specific fuel consumption, boost pressure. The experimental study was preceded by a literature analysis of the influence of the pressure drop of the inlet system, mainly the air filter pressure drop, on the engine filling and operating parameters. The constructional possibilities of reducing the air filter pressure drop and increasing the filling and power of the internal combustion engine were analyzed.

2. Literature Analysis on the Influence of the Inlet System Pressure Drop on the Engine Operation

2.1. Factors Affecting the Value of the Engine Filling Factor

The effective power of a reciprocating internal combustion engine can be expressed by the relation [36]:

$$Ne = W_d \cdot \rho_{pow} \cdot V_s \cdot i \cdot n L_t \cdot \kappa \cdot \eta_i \lambda \cdot \eta_m \cdot \eta_v, Ne = W_d \cdot \rho_{pow} \cdot V_s \cdot i \cdot n L_t \cdot \kappa \cdot \eta_i \lambda \cdot \eta_m \cdot \eta_v,$$

(1)

where: W_d —fuel calorific value, ρ_{pow} —air density, V_s —cylinder displacement, i —number of cylinders, n —engine speed, L_t —theoretical air demand, κ —stroke number factor, λ —excess air factor, η_i —indexed efficiency, η_m —mechanical efficiency, η_v —filling factor.

From the above expression, it follows that the effective power of the reciprocating internal combustion engine depends on many factors, and mainly on the filling factor η_v , which is determined from the relation:

$$\eta_v = \dot{m}'_{rz} \dot{m}'_t, \eta_v = \dot{m}'_{rz} \dot{m}'_t,$$

(2)

where: $\dot{m}'_{rz}$ —mean real flow rate of air supplied to engine cylinders in determined operating conditions and in time interval long enough to eliminate the influence of pressure pulsations in intake duct, $\dot{m}'_t$ —theoretical mass flow rate of air supplied to engine cylinders in determined operating conditions.

The value of the engine filling factor is influenced by the following factors: thermodynamic, structural and operational.

The thermodynamic factors of the charge in the cylinder are related to each other by the relation [37]:

$$\eta_v = T_H p_H (\epsilon - 1) \cdot (\epsilon p_N T_N - p_r T_r), \eta_v = T_H p_H (\epsilon - 1) \cdot (\epsilon p_N T_N - p_r T_r),$$

(3)

where: p_H , T_H —ambient pressure and temperature, p_N , T_N —pressure and temperature at the end of the filling stroke, p_r , T_r —pressure and temperature of the rest of the exhaust gas, ϵ —compression ratio.

The various thermodynamic parameters have different and not always unambiguous effects on the value of the filling ratio. However, the influence of various factors on the value of the filling ratio according to relation (3) does not lead to unambiguous conclusions.

An increase in the ambient temperature T_H causes an increase in the temperature T_N at the end of the filling stroke, which in the presence of a constant ambient pressure results in a decrease in air density and thus a lower mass filling of the engine cylinders.

The ambient pressure p_H has a significant effect on the filling ratio. This is clearly visible as the altitude at which the engine operates increases, which is associated with a decrease in air density, as well as with a decrease in ambient temperature [37]. For example, at an altitude of 1000 m above sea level, the pressure is $p_H = 893$ hPa and is lower than the pressure at sea level (1013.25 hPa) by 11%, and the ambient temperature T_H by 2.4%, resulting in a decrease in air density ρ_p by nearly 11%. The filling of the engine cylinders also deteriorates by this amount.

According to the authors of [37,38], the filling ratio is affected more by thermodynamic parameters at the inlet than by those at the outlet, especially by the end-of-fill pressure p_N . For fixed valve timing phases, the end-fill pressure p_N depends mainly on the pressure drop of the intake system Δp_u according to the relation:

$$p_N = p_H - \Delta p_u.$$

(4)

Among the constructional factors, the pressure drop of the intake system has the greatest influence on the filling factor of the internal combustion engine, which depends on:

(1)

Air filter pressure drop Δp_f —its value depends on the type of filter (baffle, cyclone) and the degree of dust contamination of the filter element,

(2)

Pressure drop of external inlet ducts Δp_D —its value depends on the cross-sectional area and the quality (smoothness) of internal duct walls, the number and radii of bends in the air path,

(3)

Pressure drop of the assembly “intake seat—intake valve” Δp_{GK} —located in the head channel, and its value will depend primarily on the lift of the intake valve h_v , which is assigned by the design angle of rotation of the crankshaft α . The value of the pressure drop of the assembly “intake seat—intake valve” can be described by the function $\Delta p_G = f(\alpha)$.

(4)

Pressure drop of carburetor and throttle Δp_G —its value depends on the opening angle of the mixture throttle. Then the above relation takes the form:

$$p_N = p_H - (\Delta p_f + \Delta p_D + \Delta p_{GK} + \Delta p_G). \quad (5)$$

For a given engine construction (valve timing, valve lift, piston speed, cylinder capacity, compressor), at a constant speed and load, the maximum filling of the engine cylinders will depend mainly on the air filter pressure drop p_f , which increases its value during the vehicle operation (**Figure 1**). The air filter pressure drop Δp_f , which increases with the operating conditions as a result of accumulation and deposition of pollutants on the filter cartridge, will cause a drop in pressure p_N at the end of the filling stroke, and thus in the filling level. As a consequence, the mass of air delivered to the engine cylinders decreases. In naturally aspirated compression ignition engines equipped with a classic in-line injection pump, there is a decrease in the excess air coefficient λ (at the same fuel dose $G_e = \text{const}$). For the compression ignition engines meeting the requirements not higher than EURO II operating in the rated conditions, the coefficient should be equal to $\lambda \cong 1.4$ [39], whereas for the engines meeting the standards above EURO II, it is necessary to increase the value of the coefficient to the value above 1.5–1.6 due to the necessity of reducing the particulate matter emission [40]. At this value of the coefficient λ in the cylinders, there are good conditions for the preparation of the mixture, the initiation of the ignition, the course of combustion and the release of heat, and thus the power achieved by the engine is the highest. Reduced air mass (oxygen deficiency in the combustion chamber) causes the fuel not to burn completely, which is one of the reasons for the formation of complete and incomplete fuel combustion products. These are primarily: PM (particulate matter) and CO (carbon monoxide), HC (hydrocarbons), NO_x, (nitrogen oxides) which are toxic components of the exhaust gas. There is a decrease in the efficiency of the engine and thus its torque M_e and power N_e .

The most important operational factors affecting the engine filling are the engine speed and its load, as well as the air pressure drop p_f increasing during operation. In most of the currently operated reciprocating engines, an increase and then a decrease in the filling ratio η_v is observed as the engine speed n increases. A typical waveform $\eta_v = f(n)$ for a naturally aspirated spark-ignition and compression-ignition engine is shown in **Figure 3**.

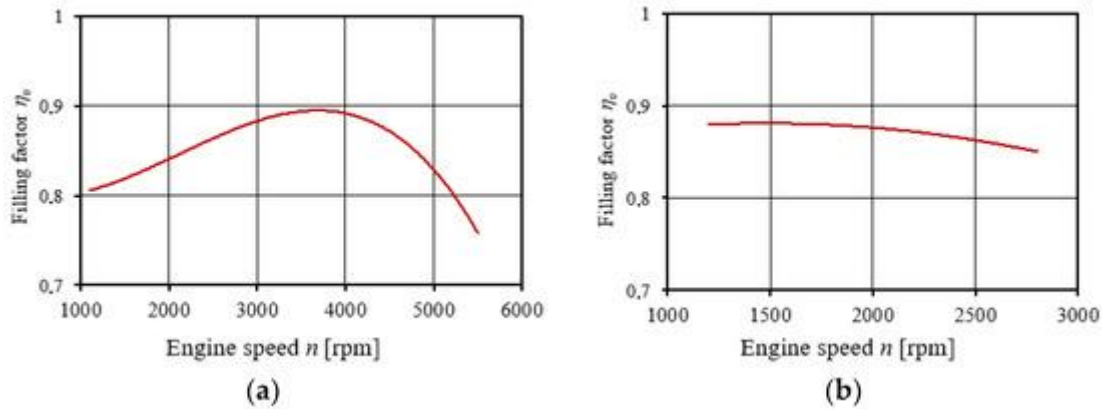


Figure 3. Dependence of the filling ratio on the rotational speed n for naturally aspirated engines: (a) spark-ignition, (b) compression-ignition.

Increasing of the filling level with increasing engine rotational speed of the SI engine results from high kinetic energy of the flowing charge and dynamic supercharging. Decreasing filling with further increase of engine speed occurs due to increasing pressure drop of the intake system with increasing speed of the flowing medium according to the relation:

$$\Delta p_U = \xi \rho_L \cdot v_{2L}^2, \Delta p_U = \xi \rho_L \cdot v_{L2}^2, \quad (6)$$

where: ξ —dimensionless pressure drop coefficient of the inlet system, ρ_L —charge density, v_L —average charge velocity in the inlet system.

The influence of the engine load on the filling ratio varies and depends on the way the fuel supply equipment is regulated. In the case of an SI engine, as the engine load decreases and the throttle is closed,

the resistance of the inlet system increases. This results in a decrease in the filling level and thus a decrease in engine power.

2.2. Influence of Air Pressure Drop on Performance of an Internal Combustion Engine

In the available literature, the dependencies, which unambiguously define the nature of the changes in the decrease of engine power depending on the increasing resistance of the air filter, are not very often found. The results of experimental research on carburetor or compression-ignition engines with classical injection systems—with a piston (sectional) in-line injection pump—are the most common [41,42,43,44,45,46,47,48].

The characteristics of the filling ratio $\eta_v = f(n)$, power $N_e = f(n)$ and torque $M_o = f(n)$ of the eight-cylinder naturally aspirated ($V_{ss} = 6.842 \text{ dm}^3$) 359M compression-ignition engine with the classical injection system are presented in Figure 4, Figure 5 and Figure 6, for three different values of the air filter pressure drop Δp_f [41]. At engine speed $n = 2800 \text{ rpm}$, the filter pressure drop had the following values:

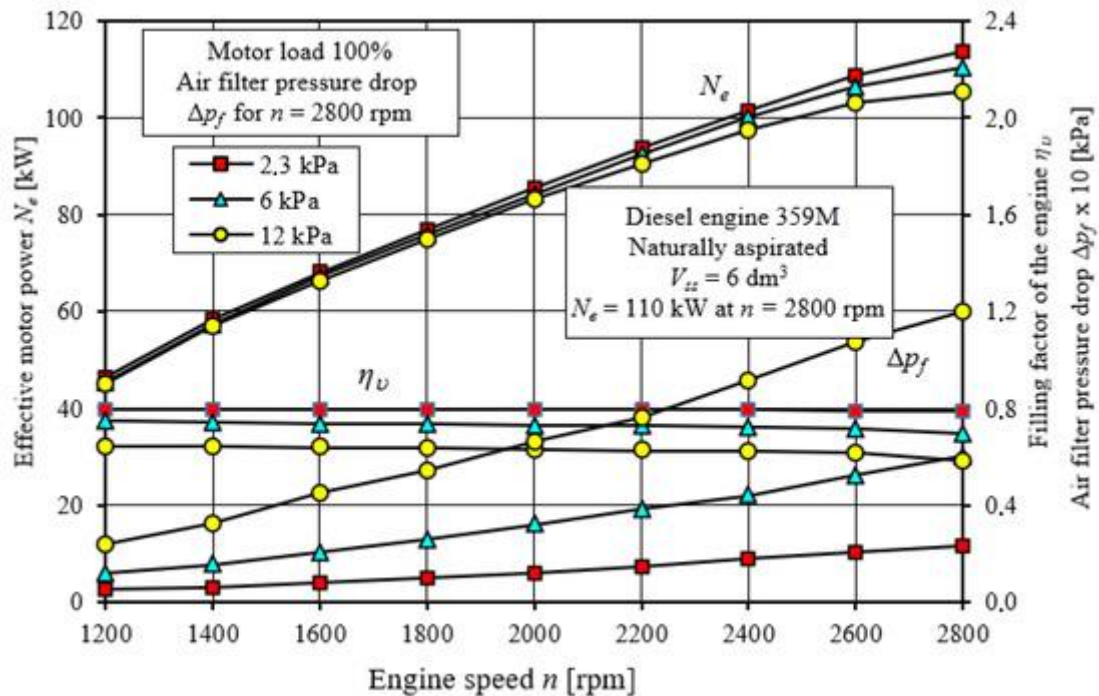


Figure 4. Dependence of fill factor and power of a naturally aspirated 359M diesel engine with classic injection system for different values of air filter pressure drop. The figure was made by the authors based on data from the paper [41].

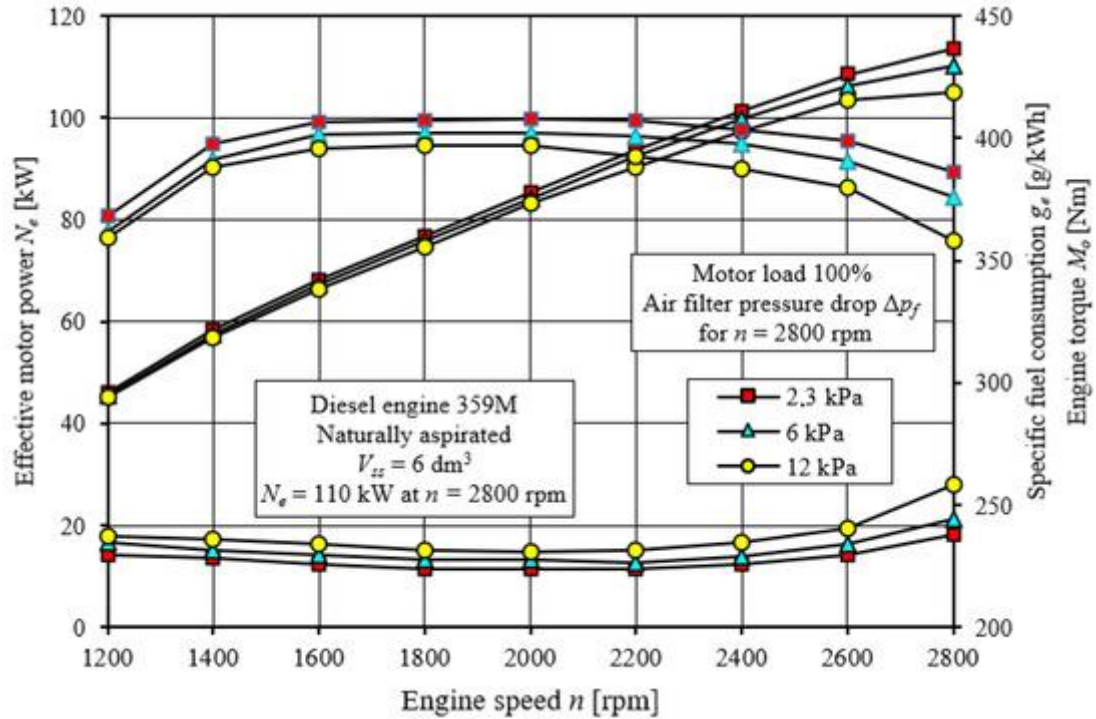


Figure 5. External characteristics of a naturally aspirated 359M compression ignition engine with a classic injection system for different air filter pressure drop (at constant speed $n = 2800$ rpm). The figure was made by the authors based on data from the paper [41].

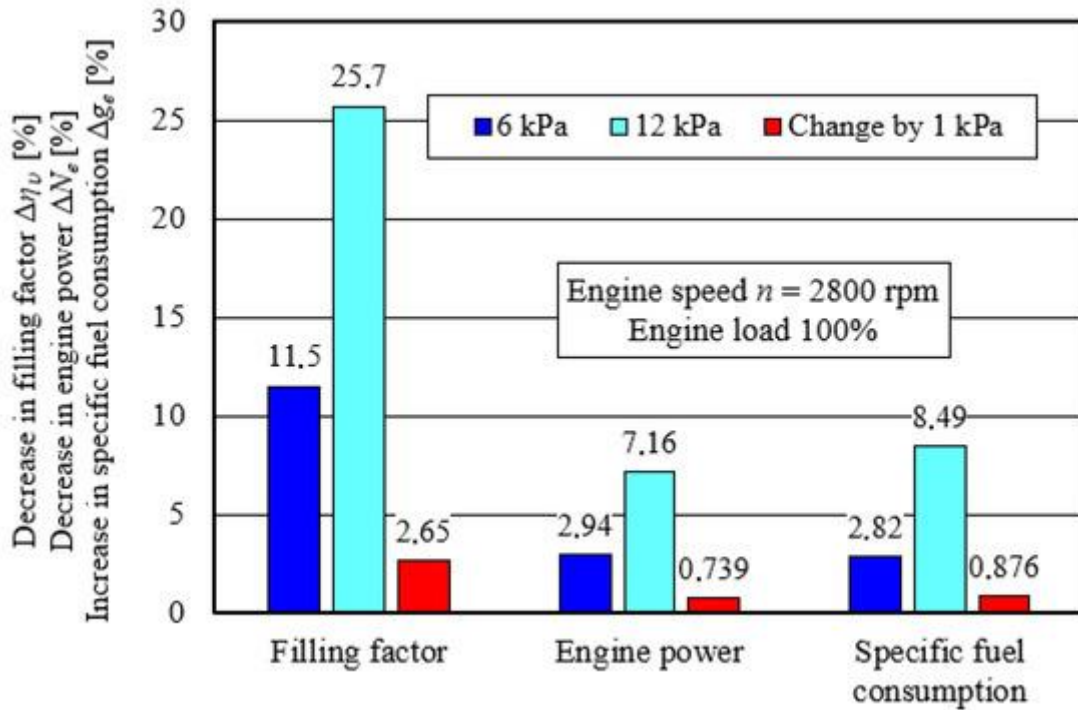


Figure 6. Influence of the air pressure drop of the naturally aspirated 359M engine with classic injection system on its operating parameters.

- air filter pressure drop with clean filter insert— $\Delta p_f = \Delta p_{f0} = 2.3$ kPa.

- permissible air filter pressure drop— $\Delta p_f = \Delta p_{fdop} = 6$ kPa,
- air filter pressure drop— $\Delta p_f = 2\Delta p_{fdop} = 12$ kPa.

As the engine speed increases, regardless of the value of the air pressure drop, for lower and medium engine speeds, the filling factor assumes almost constant values. At higher engine speeds, a slight decrease of the filling ratio η_v can be observed, resulting from the increasing resistance to the flow caused by the increasing speed of the air flow. This is, therefore, a typical characteristic curve of the filling factor $\eta_v = f(n)$ for the naturally aspirated diesel engine. As the value of the pressure drop of the air filter increases from $\Delta p_{f0} = 2.3$ kPa to $\Delta p_{fdop} = 6$ kPa, and then to $\Delta p_f = 2\Delta p_{fdop} = 12$ kPa, the filling characteristic $\eta_v = f(n)$ shifts almost in parallel towards lower values η_v (Figure 4). At engine speed $n = 2800$ rpm, the filling factor takes the values: $\eta_v = 0.785, 0.695, 0.583$. An increase in air pressure drop by 1 kPa results in a decrease in the fill factor by 2.65% on average.

The decrease in the filling ratio causes a deterioration of the engine performance: a decrease in torque M_o and power N_e and an increase in specific fuel consumption g_e . The family of curves $M_o = f(n)$, $N_e = f(n)$ and $g_e = f(n)$ for pressure drop values: $\Delta p_{f0} = 2.3$ kPa, $\Delta p_{fdop} = 6$ kPa, $2\Delta p_{fdop} = 12$ kPa are shown in Figure 5.

An increase in air filter pressure drop in the range of 2.3–12 kPa, during engine operation at engine speed $n = 2800$ rpm and 100% load, causes a decrease in the filling factor by 25.7%, in power by 7.16%, and an increase in specific fuel consumption by 8.49%. An increase in the air pressure drop by 1 kPa causes an average decrease in the filling factor by 2.65%, a decrease in power by 0.739% and an increase in specific fuel consumption by 0.876% (Figure 6).

Reducing the coefficient η_v , and thus of the mass of air supplied to the engine cylinders at the same fuel dose ($n = \text{const}$, $G_e = \text{const}$.) resulted in the decrease of the excess air coefficient λ from 1.36 at $\Delta p_{f0} = 2.3$ kPa to $\lambda = 0.96$ (at $\Delta p_f = 12$ kPa). For the diesel engine working in the rated conditions (in such conditions the tested engine worked), this coefficient should have the value $\lambda = 1.4$ [39]. At this value of the coefficient in the engine cylinders there are good conditions of the mixture preparation, ignition initiation, combustion process and heat release, and thus the engine power is the highest. Increase of the air pressure drop disturbs the mentioned phenomena. Lack of air causes the fuel not to burn completely, resulting in a decrease in engine efficiency, and consequently a decrease in torque M_o and power N_e .

Figure 7 and Figure 8 show the results of the study of the effect of the air filter pressure drop Δp_f on the external characteristics of the effective power N_e and specific fuel consumption g_e of the diesel engine of a special vehicle [42].

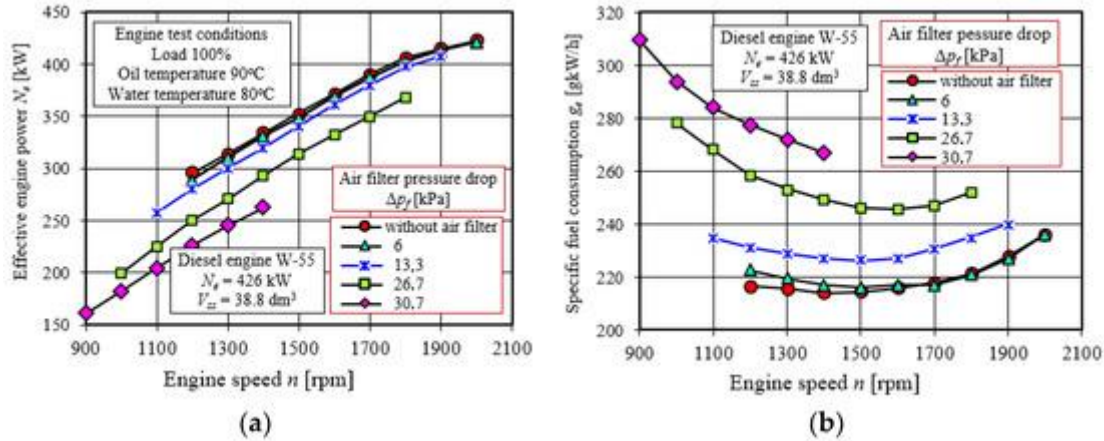


Figure 7. Influence of air filter pressure drop Δp_f on the characteristics of: (a) specific fuel consumption, (b) effective power of a twelve-cylinder, naturally aspirated W-55 compression ignition engine. The figure was made by the authors based on data from the paper [42].

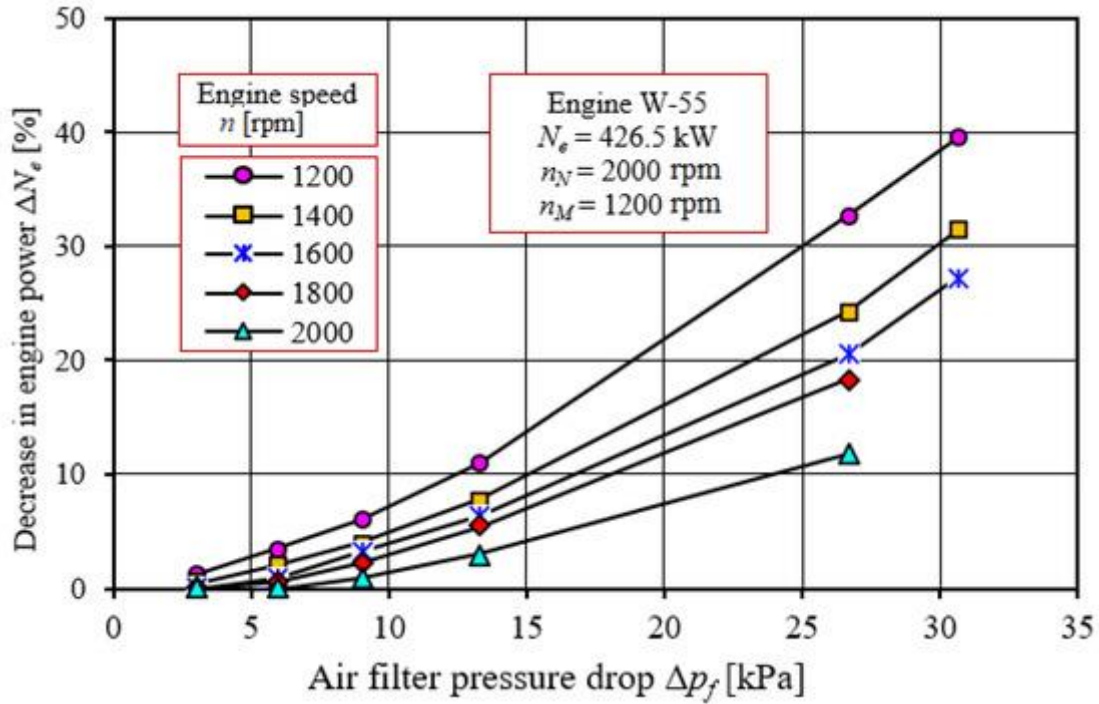


Figure 8. The effect of air filter pressure drop Δp_f on the power drop of a naturally aspirated diesel engine W-55 different speeds.

This was a 12-cylinder (V-system) naturally aspirated engine with a displacement of 38.88 dm³ and a rated power of 430 kW (580 hp) at $n = 2000$ rpm, with a classical injection system and a multi-range speed controller (direct fuel injection, 4 valves per cylinder). It can be seen from the presented graphs that the effect of pressure drop is only visible when $\Delta p_f = 6$ kPa is reached. Further increase of the air pressure drop already causes a significant decrease in engine power and an increase in specific fuel consumption, as well as a parallel shift of the external characteristics of power and specific fuel consumption towards lower values of N_e power, as well as fuel consumption g_e , with a simultaneous shift towards lower rotational speeds. At pressure drop $\Delta p_f = 26.7$ kPa, the engine operates in the speed range 1000–1800 rpm [42].

As the air filter pressure drop Δp_f increases (at a constant speed on the external characteristic curve), the percentage decreases in ΔN_e engine power increase parabolically. The decrease in power is greater the lower the motor speed is. For pressure drop $\Delta p_f = 26.7$ kPa, the power drops ΔN_e take the values: 11.75% at 2000 rpm and 20.6% at $n = 1400$ rpm and 32.7% for 1200 rpm (Figure 8).

The paper [43] presents an experimental study of the effect of air pressure drop Δp_f on the filling factor η_v and smoke opacity of a turbocharged, six-cylinder ($V_{ss} = 6$ dm³), diesel engine T359E with a classical injection system.

The tested engine, being a driving unit of a truck, was equipped with an air filter with a cylindrical paper cartridge. The effect of four technical states of the air filter, differing in pressure drop, was studied. State number one— $\Delta p_f = 3.1$ kPa—filter with a clean paper insert. Number two, three, four— $\Delta p_f = 11, 18.7, 24.7$ kPa, respectively (Figure 9). The pressure drop values were due to the different contamination state of the three paper filter cartridges.

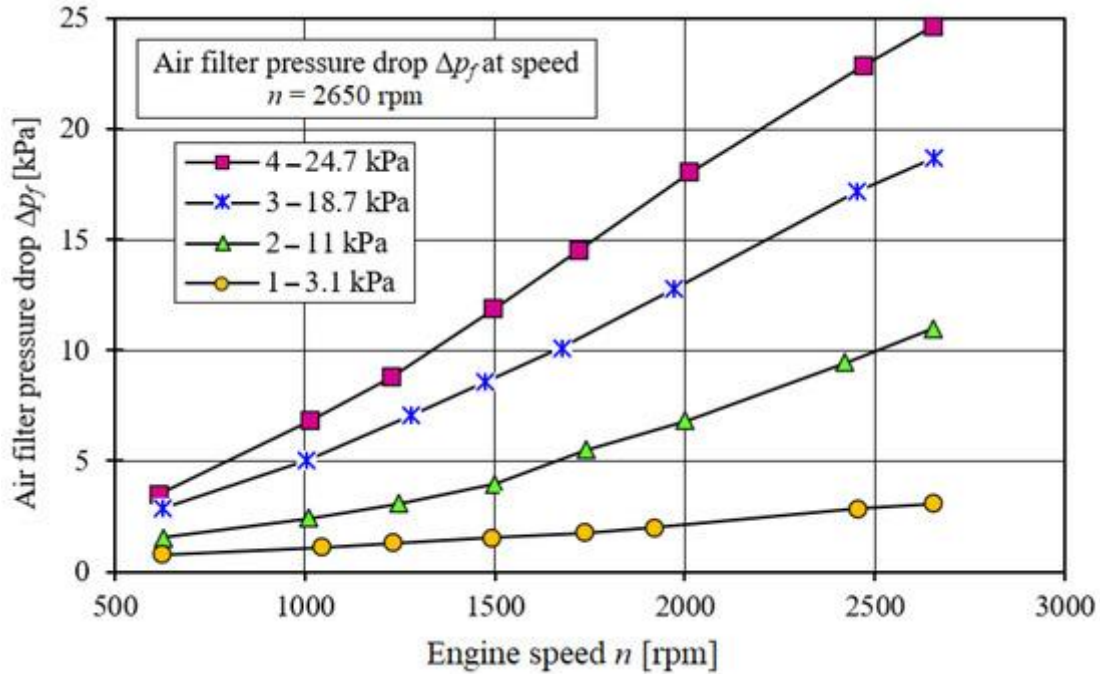


Figure 9. Flow characteristics $\Delta p_f = f(n)$ for different air filter conditions in the engine speed range $n = 650$ – 2650 rpm and without external engine load [43].

The engine filling factor η_v was determined using relation (2). To measure the real mass flow rate of the air m_{rz} supplied to the engine cylinders, the HFAM-1000 flowmeter (with hot wire) with the measurement range of 10–1200 kg/h and measurement accuracy of 5% was used. For measurement of the engine smoke opacity, the MDO-2 absorption opacimeter was used, with the measuring range of the light absorption coefficient of 0–10 m^{-1} and accuracy of 0.01 m^{-1} .

The tests were carried out using the method of free acceleration of the engine loaded with its own resistance moment and its total moment of inertia, hereinafter referred to as the “dynamic characteristics” method [44]. This method enables the determination of the instantaneous value of the engine torque without loading it on the dynamometer bench. The determination of this engine parameter consists of the precise determination of the course of the crankshaft angular accelerations during the engine acceleration process, resulting from the step change of the fuel supply to its cylinders.

The effect of the air pressure drop Δp_f on the variations of the air flow rate supplied to the engine cylinders is shown in Figure 10. The maximum value of the air stream (at the end of the acceleration process) is about 685 kg/h for the air filter pressure drop $\Delta p_f = 3.1$ kPa. For the remaining states (two, three, four) of the air filter, the air flux values were observed to be respectively smaller for each state. For state number four, the air flux $m_{pow} = 500$ kg/h. This is 25% less than for the air filter with a clean filter cartridge.

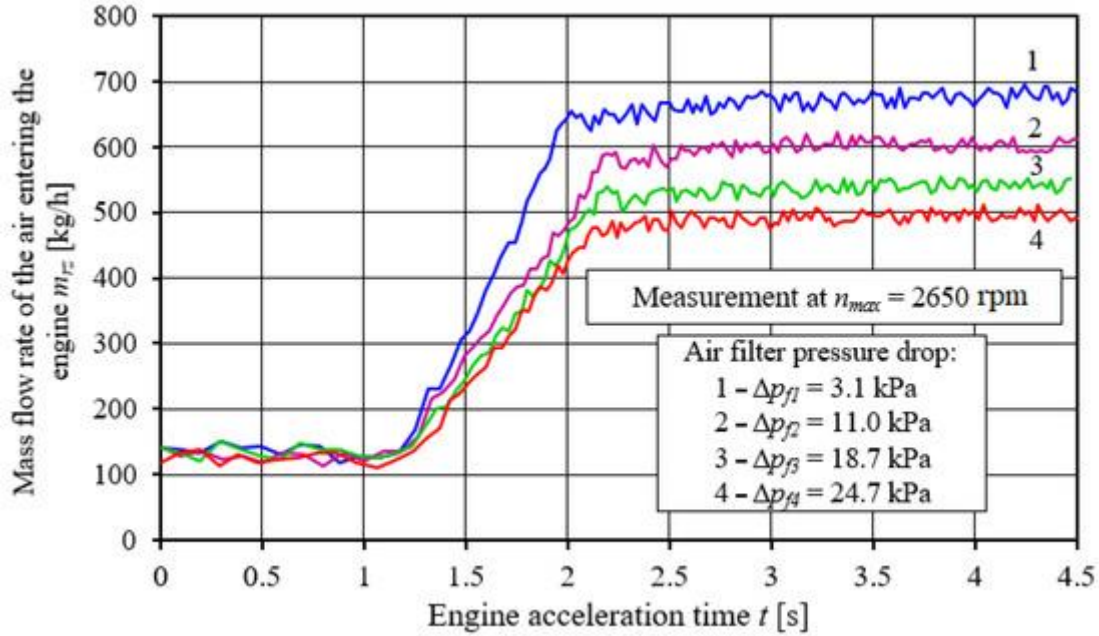


Figure 10. Mass flow rate of air m_{rz} entering the engine at different air filter condition as a function of time t of engine acceleration. The figure was made by the authors based on data from the paper [43].

The decrease in the amount of air supplied to the engine cylinders primarily results in a decrease in the filling ratio η_v , and thus in power output. The changes of the filling degree η_v for the recorded values of the air flow m_{rz} are shown in Figure 11. At the final stage of the engine acceleration process for the air filter with a clean filter insert (state number one) the filling degree is $\eta_v \approx 1.02$. With the increase of the pressure drop the filling degree takes on smaller and smaller values, respectively: $\eta_v \approx 0.90$; 0.81 ; 0.75 —Figure 11. Thus, an increase in the resistance Δp_f by 1 kPa causes a decrease in the filling degree by 1.49, 1.29 and 1.23%, on average.

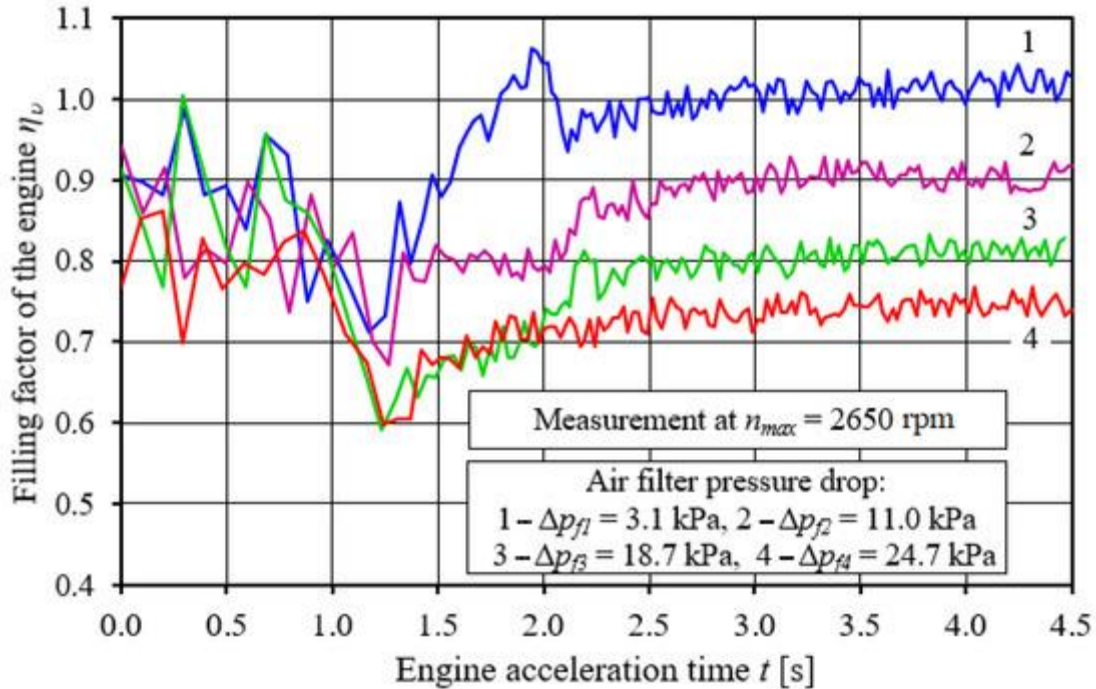


Figure 11. Changes in fill factor η_v as a function of time t of acceleration of the T359E engine at different values of air filter pressure drop—filter element contamination degree. The figure was made by the authors based on data from the paper [43].

Decreasing the filling degree, causes at the same fuel dose a decrease in the value of the excess air coefficient λ . It causes disturbance in engine work. It causes deterioration of conditions of mixture preparation, ignition initiation, combustion process and heat release, and thus decrease of engine efficiency, its torque and power.

An eight-fold increase in the resistance Δp_f over the initial resistance Δp_{f0} results in a two-fold increase in the smoke opacity of the T359E engine (increase in the light absorption coefficient) to $k = 0.81 \text{ m}^{-1}$. For the T359E engine, this does not cause exceeding its maximum value— $k_{\max} = 3.0 \text{ m}^{-1}$.

The paper [45] presents fuel consumption and exhaust emissions during operation of a carburetor engine with and without an air filter (Figure 12). When the engine is operated with a constant load in the speed range from 1500 to 2500 rpm, the hourly fuel consumption with an air filter increases from 0.687 to 1.028 dm^3/h , i.e., by 49.6%. On the other hand, when operating the engine under the same conditions but without air filter, the hourly fuel consumption increases by only 35.2%.

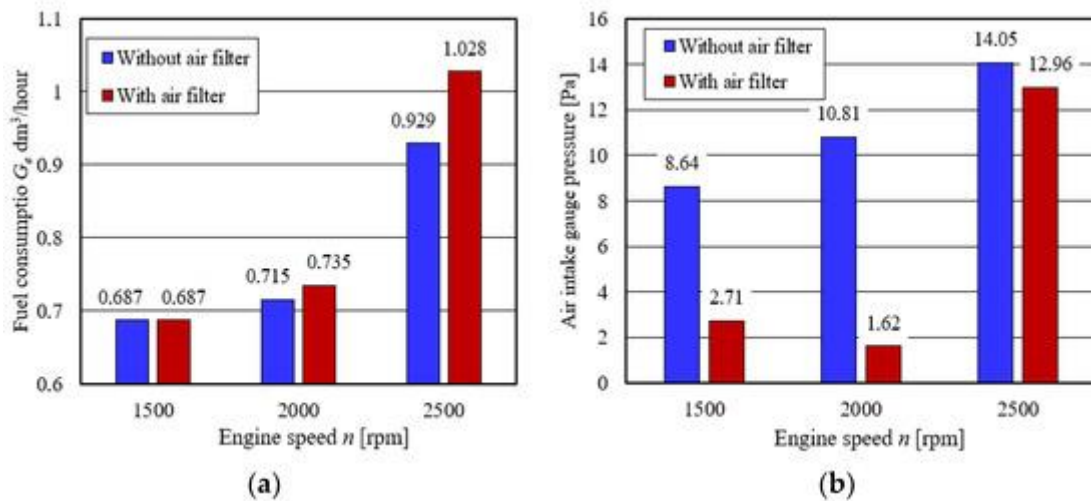


Figure 12. Effect of air filter pressure drop on: (a) fuel consumption at constant load, (b) air intake gauge pressure at constant load. Figure made by the authors using data from [45].

The air inlet pressure for the variant with air filter assumes slightly smaller values than that without air filter, especially when the engine is running at $n = 1500 \text{ rpm}$ and $n = 2000 \text{ rpm}$ —Figure 12. For the first case it is 31.2%, and in the second case it is 15% of the pressure without air filter. At $n = 2500 \text{ rpm}$ the inlet pressure with air filter has only a slightly lower value than the inlet pressure without filter. The higher inlet pressure promotes more efficient filling of the engine cylinders. Therefore, it is necessary that the air filter element be changed regularly. A clean filter cartridge guarantees a higher inlet pressure.

At the same time, the authors' study [45] shows that engine operation without an air filter is characterized by a higher volume percentage of CO_2 and NO_x in the exhaust gas. The operation of the engine with a constant load, in the rotational speed range 1500–2000 rpm (without filter), causes that the volumetric share of CO_2 in the exhaust gases being only 7.7 and 12% higher than for the operation with an air filter. For the speed of 2500 rpm it is 25% more. This is because, due to the lack of an air filter, more air mass flows into the engine cylinders. The excess air causes the fuel to burn completely and completely, producing more CO_2 as a result.

The effect of the air filter on the NO_x concentration in the engine exhaust is similar. Running the engine with a constant load at 1500, 2000 and 2500 rpm (without filter), results in a volume fraction of NO_x in the exhaust, respectively, of 51.7, 8.1 and 20.2%, higher than for during operation with air filter. The formation of NO_x in the engine is influenced by the maximum temperature and pressure of the combustion process. Without air filter the combustion process is more efficient, which leads to higher temperature and pressure of the gases produced during the process. The higher temperature and pressure in the cylinder, which are caused by more efficient combustion, promotes the formation of more NO_x .

The paper [46] presents the results of research on the effect of three variants of the baffle filter on the characteristics of the UTD-20 internal combustion engine of a special vehicle. This is a compression-ignition engine with displacement $V_{ss} = 15.8 \text{ dm}^3$ and power rating of 226 kW with a classical fuel injection system. Tests performed on a dynamometer stand included the comparison of effective parameters obtained for maximum load of the engine equipped with a standard air filter (S variant), with two other variants of the filter and engine operation:

S—standard filter, $\Delta p_{flop} = 13.2 \text{ kPa}$ —at engine speed $n = 2600 \text{ rpm}$,

A—upgraded filter (lower pressure drop by about 8 kPa), $\Delta p_{flop} = 4.9 \text{ kPa}$ —at engine speed $n = 2600 \text{ rpm}$,

B—modernized filter—operating with the engine with increased (about 7%) fuel dose.

The test results in the observations of these characteristics: external effective power N_e , torque M_e , hourly G_e and specific fuel consumption g_e , which are shown in **Figure 13**.

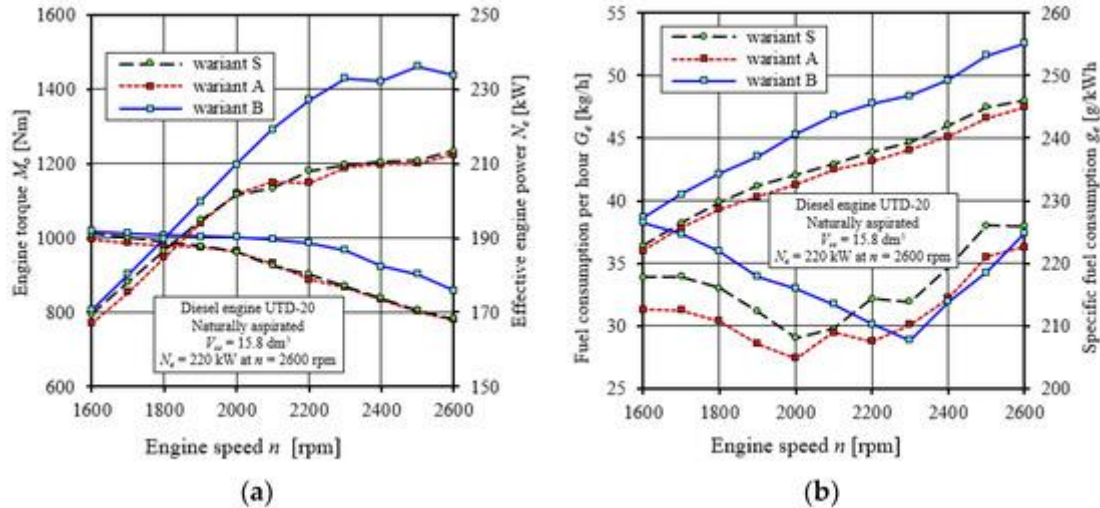


Figure 13. External characteristics of (a) effective power N_e and torque M_e , (b) hourly G_e and specific fuel consumption g_e of the UTD-20 engine. The figure was made by the authors based on data from the paper [46].

The engine working with air filter (variant A—lower pressure drop) obtained, in the whole speed range, an insignificant increase of effective parameters in comparison with the engine working with filter—variant S. The increase of power and torque is in the range of (0.4–1.2%), with lower values referring to the speed range 2300–2600 rpm. At the same time, lower (on average 1.5%) specific fuel consumption was obtained.

For the engine with an increased fuel dose and with the modernized air filter (variant B), in comparison with variant S, a significant increase in power and torque was obtained: over 2% for 1600 rpm and over 10% for 2200–2600 rpm, with a slight (2%) increase in specific fuel consumption. The increase in engine power occurred as a result of an increase in fuel delivery and increased air mass resulting from a decrease in air filter pressure drop—indicating an increase in engine filling.

According to the authors of the paper [46], an increase in the pressure drop in the intake system by an average of 10 mm Hg (1.36 kPa) causes a decrease in the power of the Jamz-240N engine by 3.68–4.41 kW and an increase in specific fuel consumption by 2.72–4.08 g/kWh. The author of the work [47] states that an increase in air pressure drop by 1 kPa causes a decrease in power and an increase in specific fuel consumption in engines with diesel engine (0.3–0.4%) and about 1.3% in engines with SI. The authors of the work [49] state that the turbine engine is more sensitive to the decrease in inlet pressure caused by the installation of air cleaning devices than reciprocating engines, and each increase in the air pressure drop at the inlet to the turbine engine by 0.5 kPa causes a decrease in its rated power by 1%.

In work [50], the effect of two types of air filter (variant A—standard filter, variant B—filter with a new structure), on the load and speed characteristics of a single-cylinder engine with a diesel engine was studied (**Figure 14** and **Figure 15**). Air filter B is much larger in size than filter A, thus has a lower pressure drop. The effective power, torque, specific fuel consumption, smoke, oil temperature and engine exhaust temperature were investigated and analyzed and compared with the parameters of engine operation without an air filter—variant W. As expected, the engine operating without air filter obtained the highest power and

torque and the lowest fuel consumption. The operation of the engine successively with the filter A and B causes a shift of the characteristics $N_e = f(n)$ i $M_o = f(n)$ almost parallel in the direction of smaller values, and the characteristics $g_e = f(n)$ in the direction of larger values, in the whole range of engine rotational speed—**Figure 14**. When the air filter is not used, the maximum torque is $M_{omax} = 75.3$ Nm at 1500 rpm. When the engine is operated with air filter type A, the maximum torque decreases to 72.5 Nm and the fuel consumption is $g_e = 234.2$ g/(kWh). With air filter type B, the torque increases to $M_{omax} = 73.6$ Nm at 1500 rpm and the fuel consumption decreases to $g_e = 230.8$ g/(kWh). This is 1.6% more torque at M_{omax} and 1.5% less fuel consumption than the engine with air filter type A.

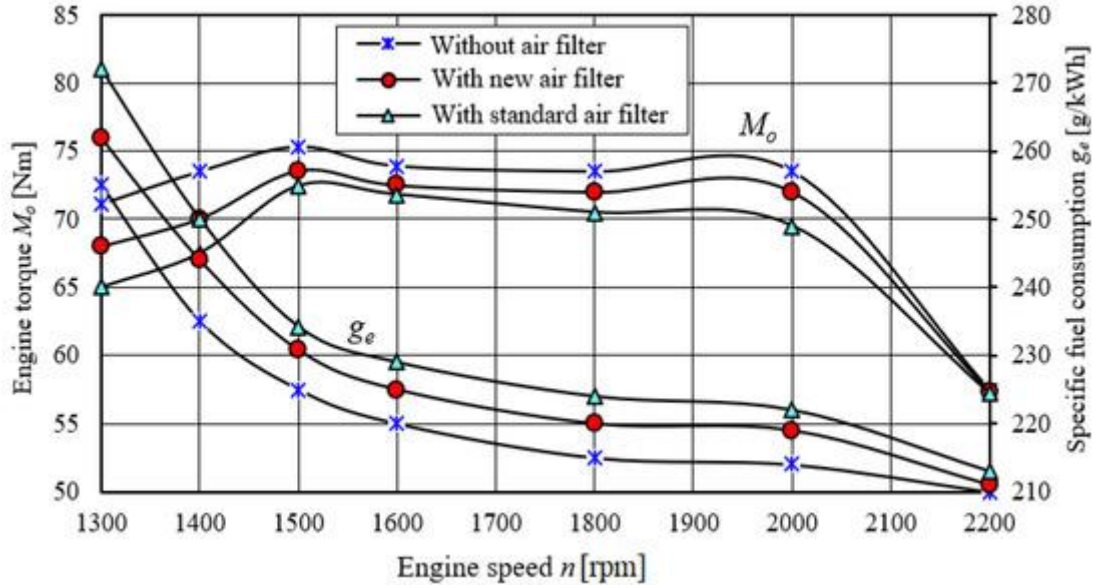


Figure 14. Torque M_o and specific fuel consumption g_e as a function of engine speed n . Figure made by the authors based on data from [50].

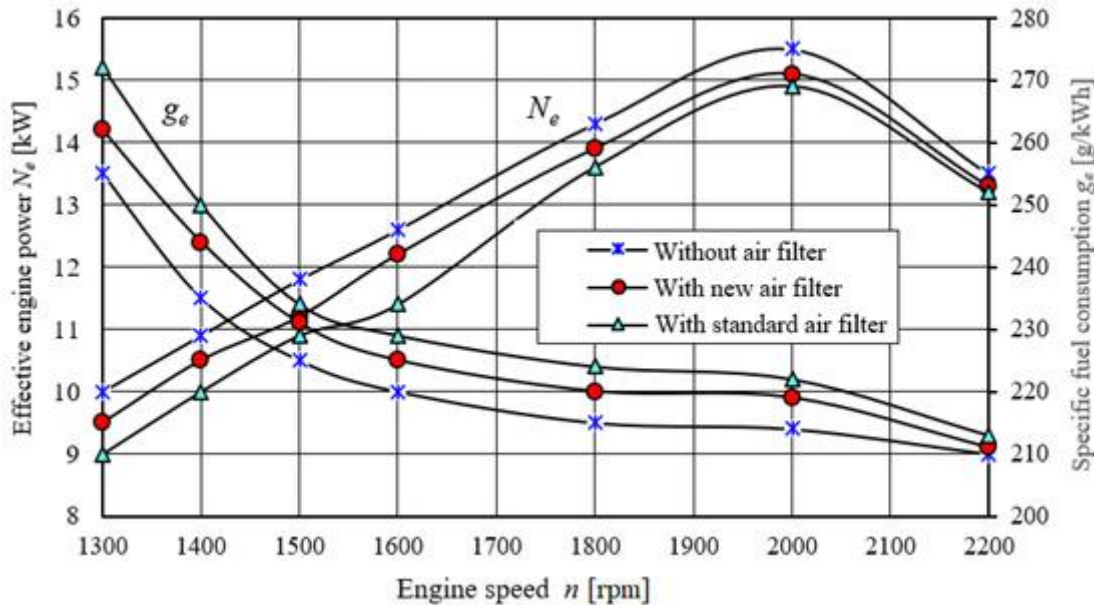


Figure 15. Effective power N_e and specific fuel consumption g_e as a function of engine speed n . Figure made by the authors based on data from [50].

The maximum power increases by 1.1% when a newly designed type B air filter is used when compared to type A air filter. The maximum smoke value (light absorption coefficient) during engine

operation without air filter, with air filter type A and with air filter type B is: $k = 1.7, 2.36$ and 2.0 m^{-1} respectively.

The paper [51] experimentally and numerically studied an air filter with a pleated filter cartridge with different parameters to improve the performance of a compression ignition engine. It was investigated what effect the paper cartridge parameters pleat height, pleat spacing, pleat shape (flat pleat, V-shaped pleat, and sinuous shape), filter paper thickness and air flow velocity have on reducing filter pressure drop. A filter media optimized for least pressure drop was designed. An experimental study was conducted on the effects of three air filters: dust contaminated filter, clean standard filter and optimized filter on engine performance. The results showed that the filter with the optimized sinusoidal air cartridge had the lowest pressure drop (Figure 16). The optimized filter also provided the lowest exhaust temperature of 218°C and the lowest fuel consumption of 290 g/kWh [51].

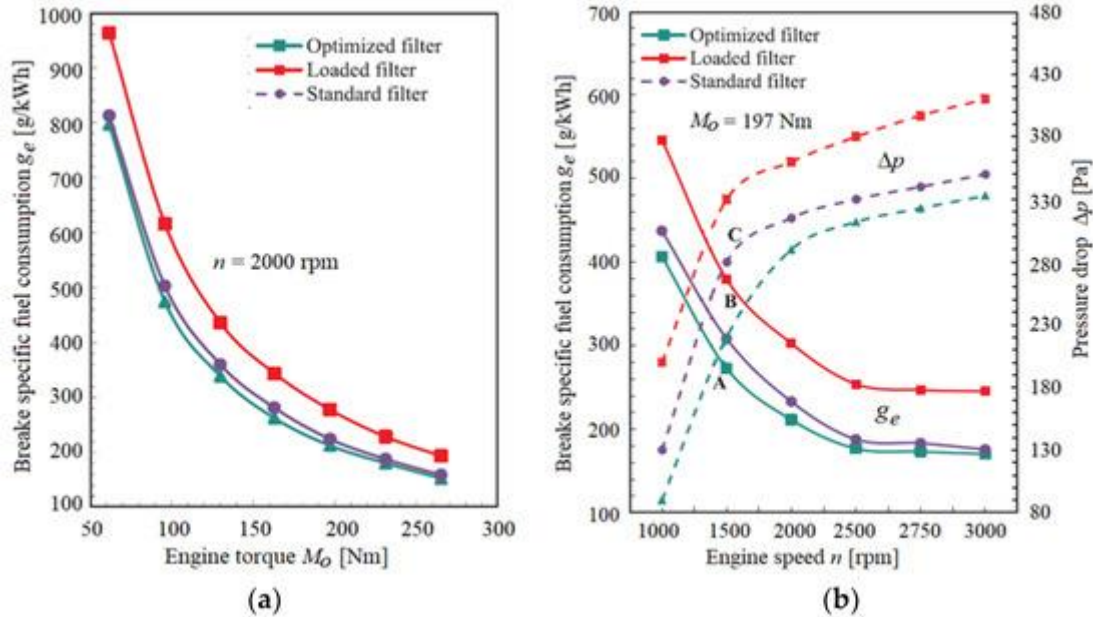


Figure 16. (a) variation of engine specific fuel consumption versus engine torque with different types of filters at constant engine speed $n = 2000 \text{ rpm}$, (b) optimization between engine specific fuel consumption and pressure drop versus engine speed with different types of filters at constant engine torque $M_o = 197 \text{ Nm}$. The figure was made by the authors based on data from the paper [51].

It was found that specific fuel consumption decreased with increasing engine torque for all filter types at constant 2000 rpm (Figure 16a). The optimized filter, due to the lowest pressure drop, provided lower fuel consumption, while the dirty filter was the cause of higher fuel consumption at the same engine torque and speed values. At a specific engine torque value of 163 Nm , the fuel consumption of the optimized filter was 262 g/kWh , while that of the standard and dirty filter was 281 and 344 g/kWh , respectively. Operating the engine with the optimized filter results in a decrease in fuel consumption when compared to the standard and dirty filter of 6.7% and 23.7% , respectively.

The intersection point of the relationship between pressure drop and specific fuel consumption, at a given engine speed and torque, was used as the optimization criterion. For the optimized pleated air filter, the recommended optimization point is located at point A (Figure 16b), where the specific fuel consumption was 290 g/kWh at a pressure drop of 205 Pa and an engine speed of 1450 rpm .

On the other hand, for the standard air filter (with pleated cartridge) and the air filter with dust contaminated cartridge, the optimization criteria were located at the intersection points B and C, for which the values of specific fuel consumption of pressure drop are higher—Figure 16b. The optimized pleated air filter with lower resistance than the others provides minimum fuel consumption at high engine speed and lower pressure loss.

The presented analysis shows that an increase in the air filter pressure drop Δp_f by 1 kPa results, on average, in a decrease in engine power by $SI (1\text{--}1.5)\%$ and an increase in specific fuel consumption by

about 0.7. For compression-ignition engines, these values are respectively a (0.4–0.6)% decrease in power and a (0.3–0.5)% increase in specific fuel consumption.

The author of the paper [52] states that each increase in the pressure drop in the air filter inlet to the compressors by 10 kPa causes an increase in energy consumption by 1%.

2.3. Possibilities for Increasing the Filling and Power of the Internal Combustion Engine

Pleated filter media show great potential for reducing pressure drop at the design stage. Lower filter pressure drop translates directly into increased fill and engine power. The great interest in this problem is manifested by a significant number of both numerical and experimental studies conducted [53,54,55,56,57,58,59,60,61]. It turned out that each pleated filter element had an optimum number of pleats, for other fixed parameters, at which it showed a minimum pressure drop.

For example, numerical studies performed by the authors of the paper [53] showed that there is an optimum number of pleats for clean filters for which the pressure drop reaches a minimum, regardless of the in-plane orientation of the fibers. At the same time, triangular pleats result in a smaller pressure drop. After accounting for particle deposition, the intensity of the increase in pressure drop decreases as the number of pleats increases. A larger number of pleats results in a higher flow velocity inside the pleat channels, which causes a greater inhomogeneity of dust deposition on the pleats. It was observed that this effect is less pronounced when the pleats are triangular in shape.

The author of this paper [54] analyzed the fluid flow through the pleated medium air filter of a four-cylinder spark ignition engine by performing experimental and computational fluid dynamics (CFD) analysis to reduce the pressure drop. It was found that the filter with triangular pleats achieves higher filtration efficiency and lower pressure drop than the filter with rectangular pleats.

For example, a numerical methodology for predicting the pressure drop in an air filter with a pleated filter element was presented in [55]. It was found that pleat geometry and inlet velocity are key parameters for filter element optimization. The optimum pleat pitch to achieve minimum pressure drop was determined and found to be significantly dependent on pleat height.

The authors of the paper [56] developed a macro-scale CFD simulation to investigate the optimal pleated air filter design under dust loading conditions. It was found that the optimum pleat density in the clean condition can lead to higher pressure drop and energy consumption during the filtration process involving dust. It was observed that the optimum dust deposition resulting from pleat density depends on the pleat height and is lower with higher pleat height.

In [57], a numerical study was presented on the pressure drop and filtration efficiency of an air filter having different numbers of rectangular and triangular pleats in both depth and surface filtration regimes with 1, 5 and 10 μm diameter particles and in the filtration velocity range of 0.5–5.0 m/s. It has been shown that filters with rectangular pleats can potentially provide better performance than their triangular counterparts at high dust loads.

The authors of the paper [58] numerically and experimentally evaluated the effect of the geometric features of pleats on the performance of fiber filters in the deep filtration phase of submicron aerosols. The paper [59] presented simulation results of the effect of the shape of pleats made of different filter materials on the pressure drop of a pleated air filter.

The paper [60] presents the study and optimization of different samples of pleated filter elements by varying the angle between pleat sides, pleat length and number of pleats. The highest filtration efficiency was obtained when the dimensionless pleat coefficient, defined as the quotient of pleat height (vertical distance from the pleat top to the base), and pleat pitch (distance between pleat tops) reached a value of 1.48. Above this value, a systematic increase in filter pressure drop was noted.

On the other hand, the authors of the paper [61] optimized the pleat geometry of automotive filters using the developed dimensionless model. Pressure drop on clean pleated filters was determined taking into account the geometric characteristics of the pleats (distance between pleats 1–3.5 mm, pleat heights 27, 32, 40, 48 mm and filtration velocity in the range 0.01–0.10 m/s). It was found that for a given pleat height and a constant air flow, there is such a width between the pleats for which the pressure drop reaches the smallest value.

The length, diameter and inner surface finish of the inlet pipes directly affect the resistance to flow of the medium. Intake pipes have a complex geometry, in part because of the space available around the engine. The geometry of the intake system affects the mass flow of air, which is hindered by interference

from pressure fluctuations generated by moving pistons and valves. In order to improve engine filling, increase engine performance and reduce fuel consumption, research work is carried out including: optimization of the geometry of the intake system components, particularly the length of the intake manifold [62,63,64,65], use of a Helmholtz resonator to generate pulsatile flow in the engine intake manifold [66,67], exploitation of airflow dynamics in the form of wave phenomena and pressure wave resonance in the intake system of an internal combustion engine resulting from cyclic opening and closing of valves [19,68,69], generation of pulsatile flow in the exhaust manifold by changing the length of the exhaust pipe [70].

The authors of the paper [71] obtained an increase in engine performance and a decrease in fuel consumption of a naturally aspirated engine by optimizing the airbox geometry. The effects of inlet diameter, airbox volume, throttle diameter and intake manifold length were analyzed. The following optimal intake system component dimensions were determined: intake diameter 81.07 mm, airbox volume, throttle body diameter 44.63 mm and intake manifold length 425 mm. This allowed for maximum engine performance (torque and power) and minimum fuel consumption.

The authors of the work [2] believe that the air flow in the intake system is disturbed by the tapering and diverging of the system components, which can lead to sudden acceleration and deceleration of the flow and cause excessive turbulence and pressure drop in the intake system, which affect the pressure drop and thus reduce the engine performance.

In [63], the effect of changing the length (volume) of the intake manifold on the performance of a ZI engine with multi-point injection system and electronically controlled fuel injectors was experimentally investigated. The results showed that changing the length of the intake manifold results in improved engine performance characteristics, especially fuel consumption at high load and low engine speeds. Therefore, a variable length intake manifold is indicated especially in urban and suburban areas (roads) where there are frequent stopping and accelerating under starting conditions. It was determined that the length of the intake manifold should be increased for low engine speeds and shortened as the engine speed increases.

Similar conclusions were reached by the authors of the paper [64], who studied on a dynamometer bench the effect of the length and diameter of the inlet pipe on the performance of the SI engine. The experiments were carried out in the engine speed range 1500–6500 rpm. Three intake pipe lengths (0.3, 0.6 and 0.9 m) and three intake pipe diameters (0.044, 0.053 and 0.067 m) were investigated on intake air mass flow rate and fill factor, torque, power, thermal efficiency and specific fuel consumption. The results showed that at low engine speeds, the intake pipe with longer length and smaller diameter provided the best engine performance. On the other hand, the intake pipe with shorter length and larger diameter provided the best engine performance at high engine speeds. The length and diameter of the intake pipe have no clear effect on specific fuel consumption and thermal efficiency of the engine.

In order to improve engine filling, the authors of [66] experimentally investigated the pulsatile flow characteristics in the engine intake manifold involving a Helmholtz resonator with variable internal volume. Using an electronic control, the internal volume of the resonator was changed and adjusted to the valve operating frequency and the intake manifold natural frequency. The results showed that the internal volume of the resonator and the frequency tuning affect the mass flow rate of the air in the intake system. For a fixed volume, the average increase in mass flow rate was 17.8%, and when the volume was tuned to the valve frequency, the average increase was 24.7%. The highest intake air mass flow rate was increased by 31.5% when the resonator was tuned to the system frequency.

The authors of the paper [19] presented the effect of variable valve timing and valve lift on internal combustion engine performance and fuel consumption. The valve lift and valve profile, which are the main factors affecting the dynamics of the gas pressure wave in the combustion chamber, were changed at all engine speeds in order to obtain an enhanced pressure wave that increases the filling of the engine cylinders. As a result of changing the above parameters, engine torque and power increased by an average of 6.02% over the entire engine speed range. In the lower speed range of 3000–4000 rpm the improvement is about 18.72%.

The paper [68] presents the performance increase obtained by fine tuning the intake system of a spark ignition engine over its entire speed range. During numerical tests, the diameter of the intake manifold and the timing phases of the intake valves were changed. When only the intake manifold diameter was changed and optimized for each engine speed, an improvement of approximately 8.5% in fill factor was obtained over the performance of the factory engine. Changing the intake valve timing phases at each engine speed resulted in a fill rate improvement of about 3%. However, combining a change in intake manifold diameter with a change in intake valve timing produced an improvement of about 12%.

In [69], the individual and combined effects of varying intake manifold length and intake valve lift on internal combustion engine performance at engine speeds from 3000 to 9000 rpm were studied. The resulting combined effect shows an average power improvement of 7.02% over the entire engine speed range.

In [70], the effect of tuning a pulsating sound wave produced over a wide range of speeds in the exhaust manifold on the power and torque of a single cylinder SI engine was analyzed. The simulation results were compared with standard data obtained during engine testing. An average of 7% increase in torque and 6% increase in horsepower was observed with constant change in tailpipe length. With a constant change in tailpipe diameter, a 6% increase in torque and power was observed. A combination of changes in exhaust pipe length and diameter resulted in 8.5% increase in torque and 9% increase in power, respectively.

A particular way to increase engine cylinder filling is to reduce the air temperature in the intake system by using an insulating layer between the engine and the intake system [71]. Insulating the intake system resulted in an increase in intake air density and an increase in fill factor of 5.88%. The mixture throttle opening with the isolated intake system required to achieve the same engine speed was reduced by about 19.8%.

From the above analysis, it can be seen that there are many feasible technologies and ways to increase engine fill and power, which boil down to reducing the pressure drop of the intake system, and this includes reducing the pressure drop of the air filter in its design phase.

It should be stated that, as of today, in the available literature there are no results of investigations of the influence of the flow resistance in the inlet system on the performance of a modern diesel engine used for driving trucks and truck tractors currently travelling on the roads and constituting the basic means of transporting goods. Therefore, it is purposeful to determine experimentally the relation between parameters of operation of the inlet system of a modern truck engine and its performance in terms of changes in emission of particular components of exhaust gases and traction properties of the vehicle.

3. Experimental Study of the Influence of the Air Filter Pressure Drop on the Performance of the Diesel Engine

3.1. Aim and Subject of Research

The objective of this study was to experimentally evaluate the effect of air filter pressure drop Δp_f on the performance parameters of a modern diesel engine with electronic control of fuel supply and supercharging and air cooling systems.

The subject of the research was a six-cylinder diesel engine of a Volvo D13C460 EURO V EEV truck with a maximum power of 338 kW, which is a power unit of a Volvo FH13 truck tractor with a mileage of 790,500 km. The total operating time of the engine was 11,800 h, and the engine consumed 229,100 dm³ of diesel fuel from the beginning of operation to the day of testing. The tested engine meets the requirements of EURO V standard according to the homologation documents. The factory external characteristics of the engine are shown in [Figure 17](#) and the basic engine parameters are shown in [Table 1](#) [72].

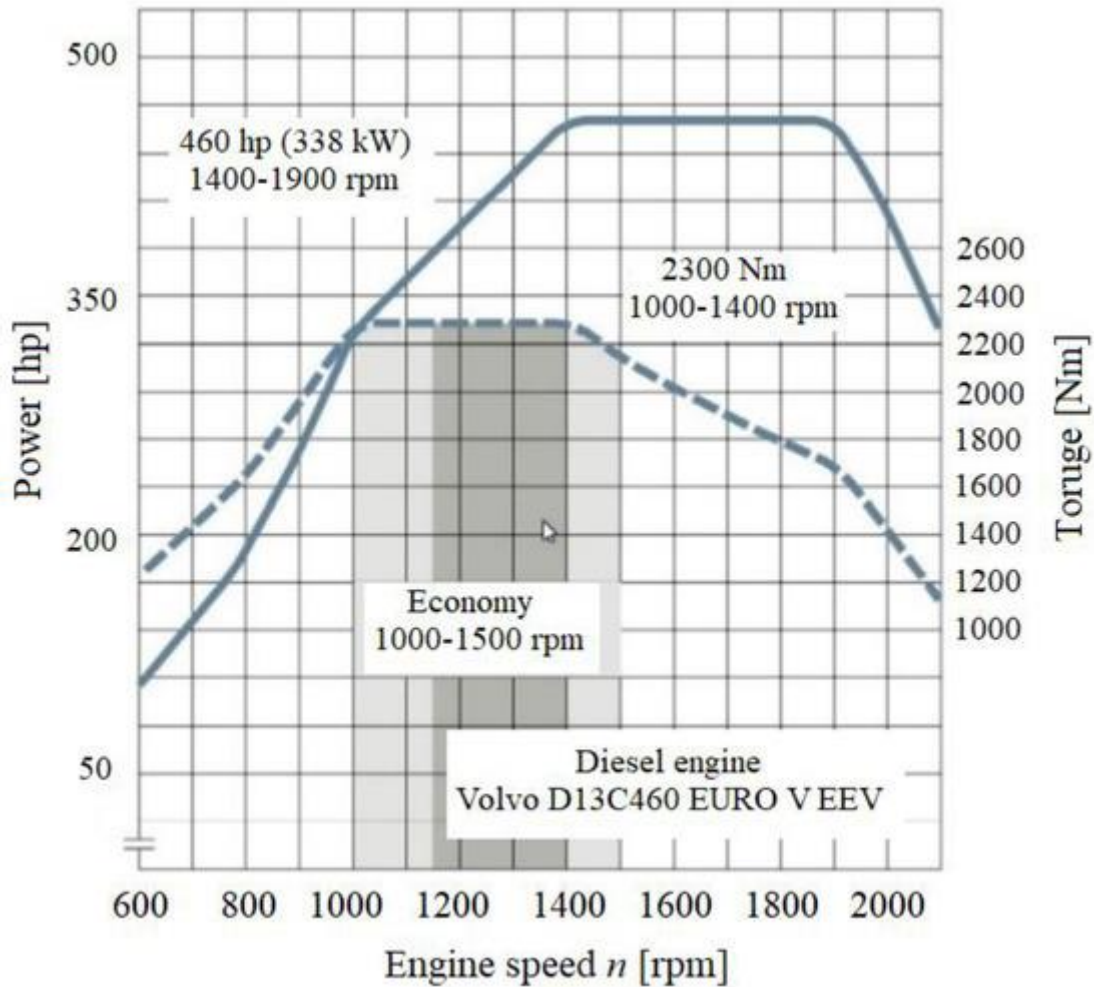


Figure 17. External characteristics of the VOVLO DC13C460 motor (338 kW) according to manufacturer specifications. The figure was made by the authors based on data from the paper [72].

Table 1. Basic parameters of the engine D13C460 EURO V [72].

A 6-cylinder, in-line, classic engine with a direct fuel injection—with an electronically controlled pump-injector power system—was studied. The engine has four valves per cylinder, which are controlled using hydraulic tappets driven from a central camshaft located on the head. This shaft also drives the electronically controlled pump injectors using a piezo-quartz valve. The valve controls the timing of the start of a specific stage of fuel injection, as well as its duration, which determines the fuel dose based on information about: intake manifold air pressure and temperature, exhaust gas temperature, ambient conditions and the desired torque resulting from the position of the accelerator pedal. The pump-injector supply system in relation to the sectional pump, shows a number of advantages. The most important include: the absence of high-pressure lines, minimal ignition delay due to the absence of delay caused by the pumping of fuel through the injection lines and high injection pressure even at minimum revolutions per minute. There are no problems with engine start, which is very important in winter conditions. The injection pressure is higher than in other systems (205 MPa), which creates good conditions for mixing fuel with air and its rapid evaporation and combustion. There is no phenomenon of injector leakage, which reduces fuel

consumption. All these features make the engine powered by a pump injector have a high efficiency (low specific fuel consumption).

The air supply system consists of an air intake located on the right side of the cabin at its highest height (**Figure 18**), an external intake duct of approximately rectangular cross-section located on the rear wall of the cabin, a rubber accordion element connecting the external duct with the air filter intake duct and a single-stage (baffle) air filter. A permissible pressure drop sensor is installed on the air filter outlet line, set at $\Delta p_{fdop} = 4.8\text{--}5.0$ kPa. Proper filling of engine cylinders is assured by turbocharger and charge air cooler operating in the “air-to-air” system. Prior to testing, the entire vehicle intake system was inspected to check for leaks and was thoroughly cleaned. This action was done to reduce the impact of unidentified damage to the intake system on the test results.

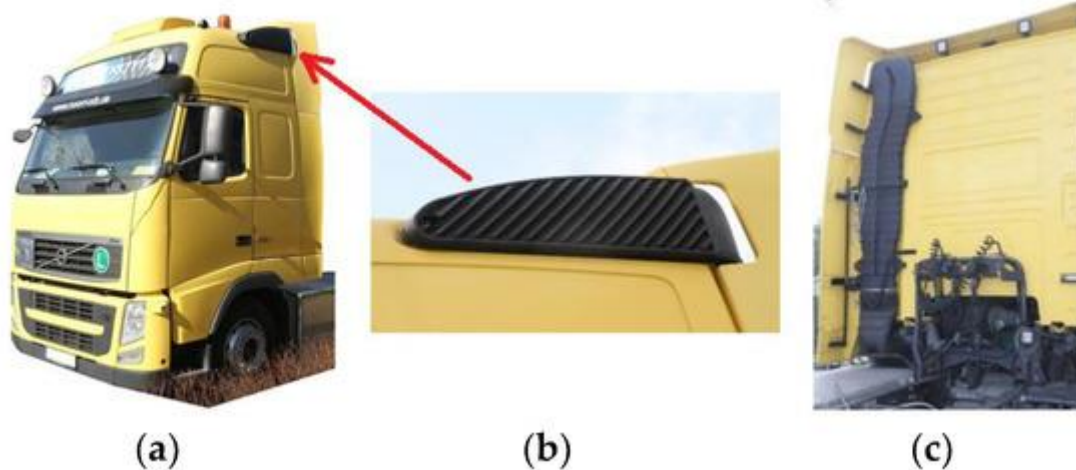


Figure 18. A Volvo FH13 truck tractor: (a) general view of the vehicle, (b) air intake, (c) view of the air supply system.

The filter element of the filter is a cylindrical cartridge made of pleated filter paper (**Figure 19**) with an active area $A_c = 13.72$ m².



Filter cartridge parameters:

Outside diameter $D = 326$ mm

Pleat height $b = 58$ mm

Active height $H = 350$ mm

Number of pleats $i_p = 338$

Filter surface $A_c = 13.72$ m²

Figure 19. Cylindrical air filter element for Volvo D13C460 EURO V engine.

3.2. Test Methodology and Conditions

Tests were carried out on a standard dynamometer bench. The motor was loaded with a water brake type Zöllner PS1-3812/AE with maximum power of 1250 kW. The torque M_o generated by the motor was measured with a strain gauge transducer connecting the swinging brake housing with the foundation. The engine speed n was recorded with a pulse transducer cooperating with a gear wheel located on the dynamometer flange. Hourly fuel consumption G_e was measured using an AVL fuel balance with a five-second time interval, and the measurement was then averaged over a 60 s interval. The coolant temperature t_{ch} during engine testing was set equal to the operating temperature of $t_{ch} = 87\text{--}92$ °C and was maintained using an external heat exchanger. The opacity of the exhaust gases was determined using an

AVL 439 OPACIMETER opacimeter, which works on the principle of measuring the light absorption coefficient. The volumetric air demand Q_s by the engine was recorded with a thermo-areometric flow transducer. The air pressure drop Δp_f was determined as the pressure difference p_1 before and p_2 after the air filter, using a TESTO 400 differential pressure gauge.

Due to the turbocharged, electronically controlled fuel injection air supply system used in the engine, it was determined that the increased air flow resistance through the filter system would have the greatest effect on the maximum power speed characteristics. The research presented in [42,45] shows that the increased flow resistance does not significantly affect the engine performance at part load. The research was conducted using the passive experiment method, by stepping the engine speed at its full load—100% accelerator pedal position.

The following methodology was used during engine testing. After establishing the thermal equilibrium conditions of the engine, the accelerator pedal was set to the position corresponding to the full engine load, and then the lowest ($n = 1000$ rpm) possible rotational speed, at which the engine was still running steadily, was set using the brake control system. After the operating conditions of the engine had stabilized, its operating parameters were recorded. The engine load was then reduced to obtain the next higher speed (in 100 rpm increments), and after the engine operating conditions stabilized, the engine operating parameters were again recorded. In the above manner, the engine operating parameters were recorded for rotational speeds in the range of 1000–2100 rpm. Due to the characteristics of the water brake used, it was not possible to perform measurements below a speed of 1000 rpm, however this did not affect the test results obtained, as speeds below 1000 rpm are not used during normal operation of vehicles with this type of engine.

During the tests, the following engine operating parameters and air parameters in the intake system were directly measured for each speed:

- engine torque, M_o [Nm],
- engine rotational speed, n [rpm],
- hourly fuel consumption, G_e [kg/h],
- engine air demand, Q_s [m³/h],
- exhaust gas temperature, t [°C],
- Smoke opacity of exhaust gases—light absorption coefficient, k [m⁻¹],
- air pressure before p_1 and after air filter p_2 , [kPa],
- charge air pressure, p_d , [kPa].

On the basis of directly measured values of engine working parameters the following were determined:

- effective engine power, N_e [kW],
- specific fuel consumption, g_e [g/(kWh)],
- air filter pressure drop Δp_f [kPa].

All tests were repeated in duplicate to eliminate coarse errors that could lead to incorrect inferences.

The control of engine load, engine speed and brake load was performed from the control and measurement cabin, where there are also indicators for measuring engine operating parameters.

During the tests, the engine operation was continuously controlled by using the diagnostic interface NAVIGATOR TXTs with the software IDC 5 TRUCK. The schematic diagram of the test stand is shown in [Figure 20](#). The applied measurement equipment and its accuracy are presented in [Table 2](#). Before and after the tests, all measurement systems important for the measurement results were checked using external standards: the exhaust gas analyzer—with standard gases, the opacimeter—with optical filters of known light absorption coefficient, the torque measuring system—with standard mass standards, the speed measuring system—with a standard tachometer, the pressure measuring systems—with standard sensors, the fuel consumption measuring system—with a dedicated mass standard.

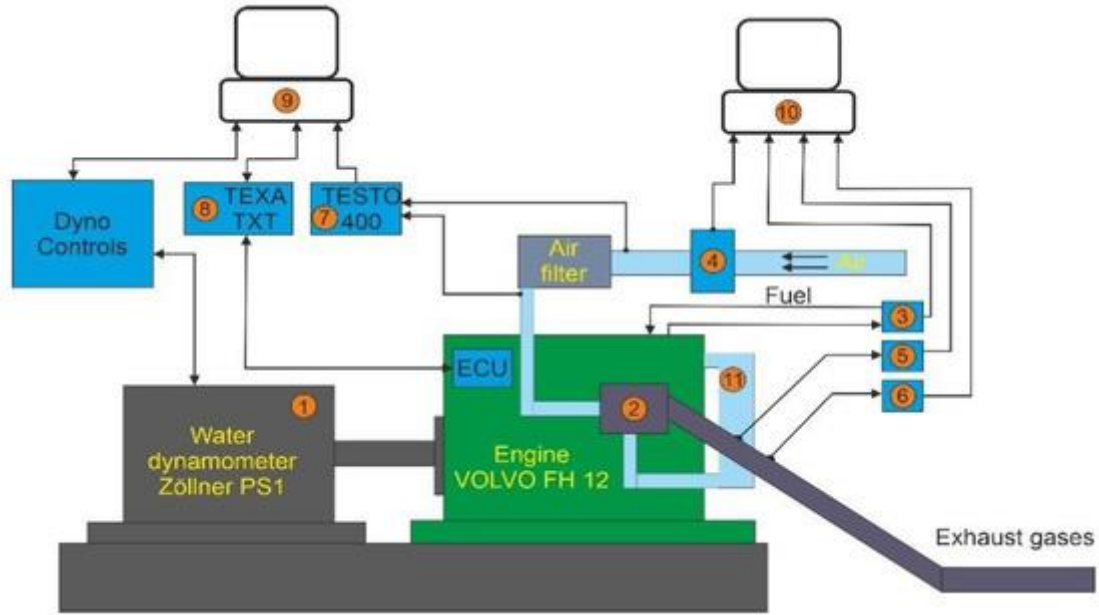


Figure 20. Diagram of the dynamometric test stand with a Volvo D13C460 EURO V engine: 1—water brake, 2—turbocharger, 3—fuel consumption measuring system, 4—air consumption measuring system, 5—smoke opacity measuring system, 6—exhaust gas temperature measuring system, 7—pressure measuring system in the engine intake system, 8—TEXA TX diagnoscope, 9—computer controlling the operation of the dynamometric brake and recording engine operating parameters, 10—computer controlling the operation of measuring systems, 11—charge air cooler.

Table 2. List of investigation equipment used during investigation.

The results of engine operating parameters: power and hourly fuel consumption, obtained during the tests, were reduced to normal conditions in accordance with the PN-ISO 15550:2009 standard [73].

3.3. Analysis of Research Results

During the experimental research the influence of four (New, A-33, B-66, C-90), differing in pressure drop, technical states of the same air filter on the external characteristics of the Volvo D13C460 EURO V engine was determined. In each case, the same parameters characterizing the engine operation were measured. After each test, the engine was inspected to detect any damage to the intake system that could adversely affect the test results.

Increase of the value of the filter pressure drop Δp_f was modeled by means of obscuring a part of the active filtration surface of the cylindrical cartridge. As a result, four technical states were obtained, differing in the value of the pressure drop of the same air filter.

- condition New—air filter with clean, brand new, paper air filter insert, $\Delta p_f = 0.58$ kPa
- condition A-33—air filter with an air filter insert that has had approx. 33% of its active filtration surface obscured, $\Delta p_f = 0.604$ kPa
- condition B-66—air filter with a filter insert, which is obscured by approx. 66% of the active filtering surface, $\Delta p_f = 0.757$ kPa
- condition C-90—air filter with a filter insert, which has approx. 90% of its active filtering surface obscured, $\Delta p_f = 2.024$ kPa.

The results of the engine tests with different air filter conditions are presented as characteristics $N_e = f(n)$, $g_e = f(n)$, $G_e = f(n)$, $p_d = f(n)$, $Q_s = f(n)$, $\Delta p_f = f(n)$, $t_s = f(n)$, $k = f(n)$ in [Figure 21](#), [Figure 22](#), [Figure 23](#), [Figure 24](#), [Figure 25](#) and [Figure 26](#).

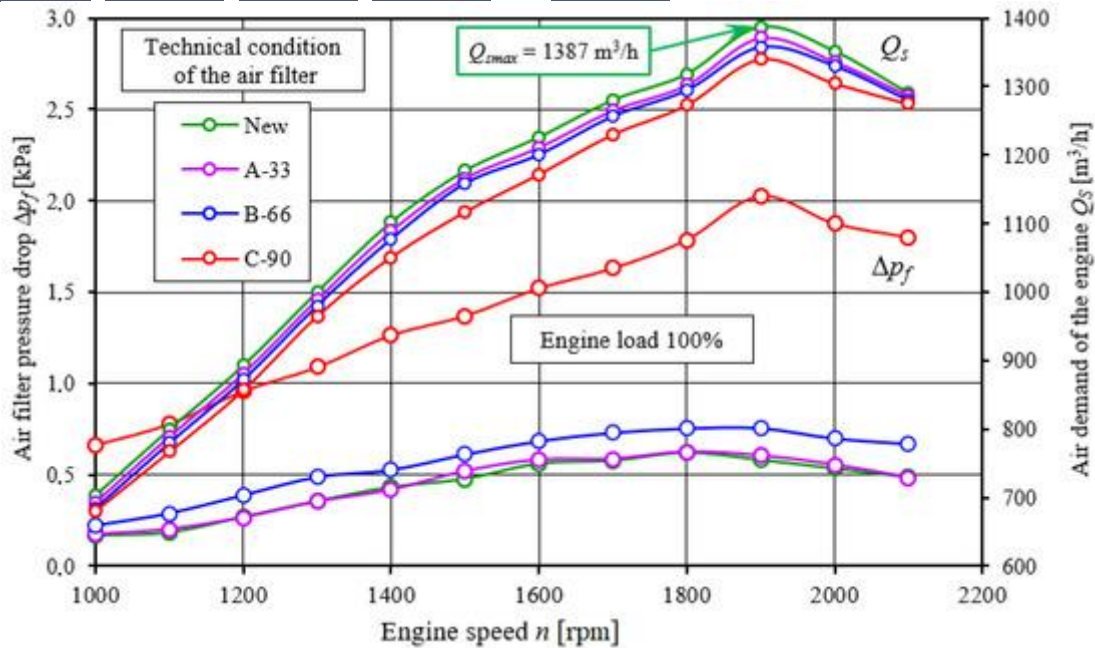


Figure 21. Pressure drop of different air filter states (New, A-33, B-66, C-90) and air demand of Volvo D13C460 EURO V engine as a function of speed.

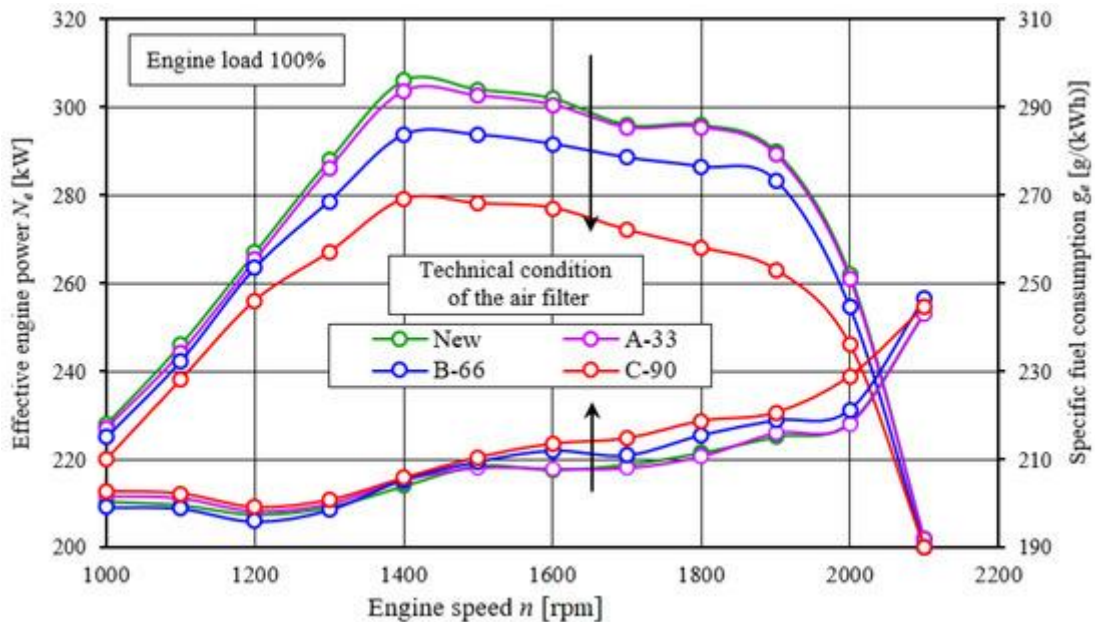


Figure 22. Effective engine power N_e and specific fuel consumption N_e of the VOVLO DC13C460 engine as a function of speed n for different air filter states New, A-33, B-66, C-90.

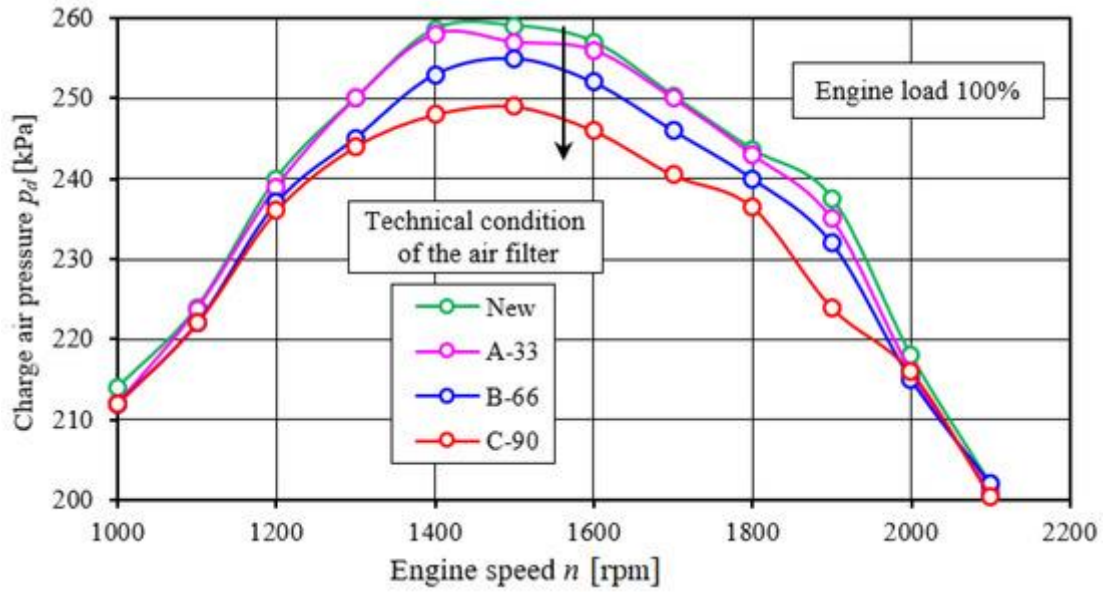


Figure 23. Charge air pressure p_d in the intake manifold of VOVLO DC13C460 engine as a function of engine speed n for different air filter states: New, A-33, B-66, C-90.

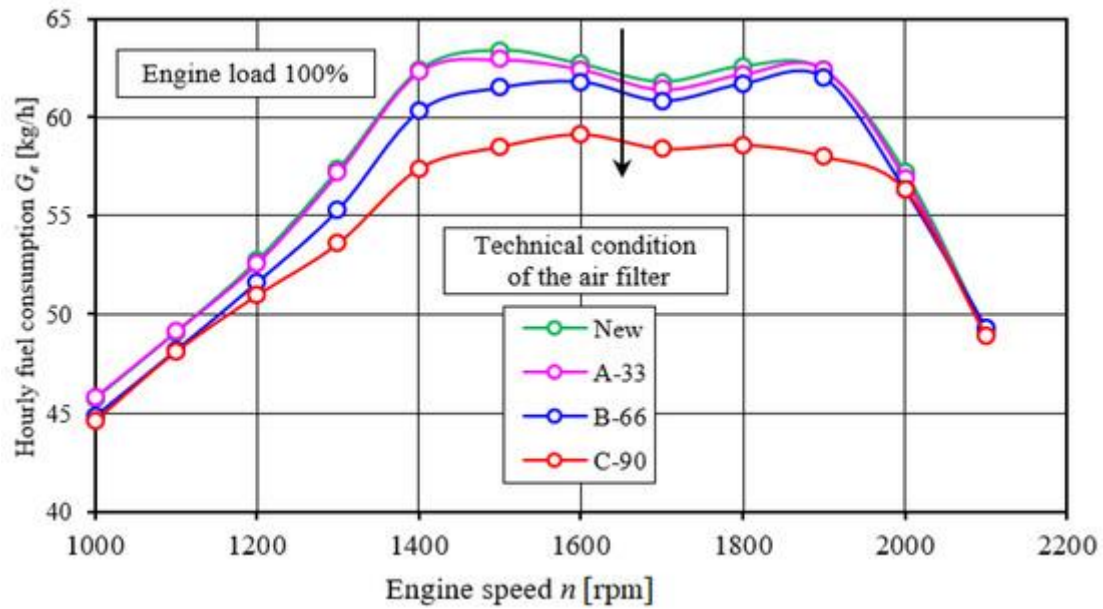


Figure 24. Hourly G_e fuel consumption of VOVLO DC13C460 engine as a function of engine speed n for different air filter conditions: New, A-33, B-66, C-90.

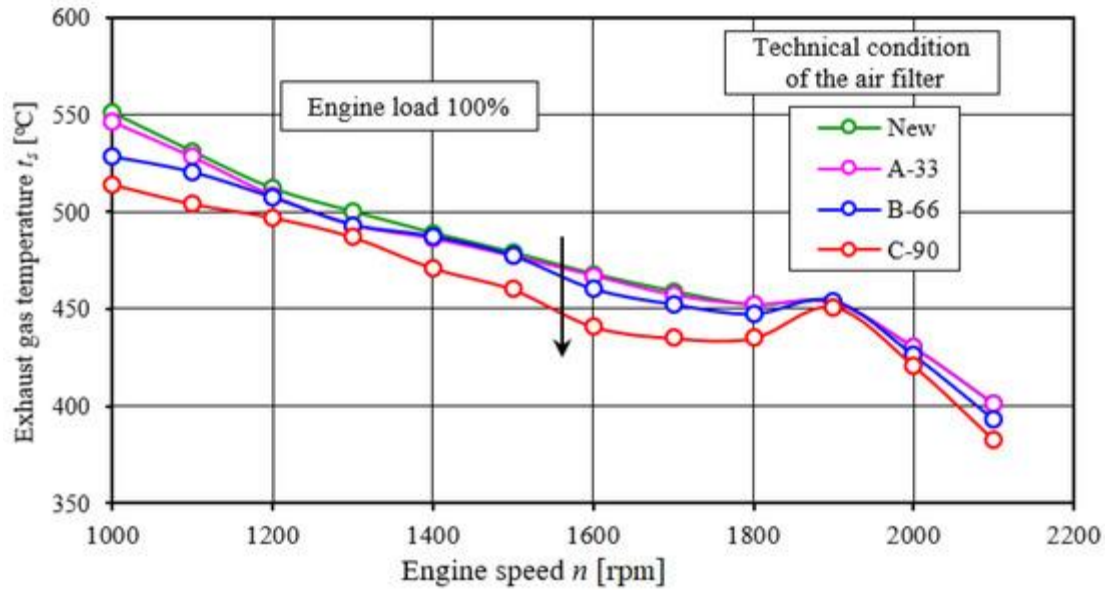


Figure 25. Exhaust gas temperature t_s at the turbine exit of the turbocharger unit of the VOVLO DC13C460 engine as a function of engine speed n for different air filter states: New, A-33, B-66, C-90.

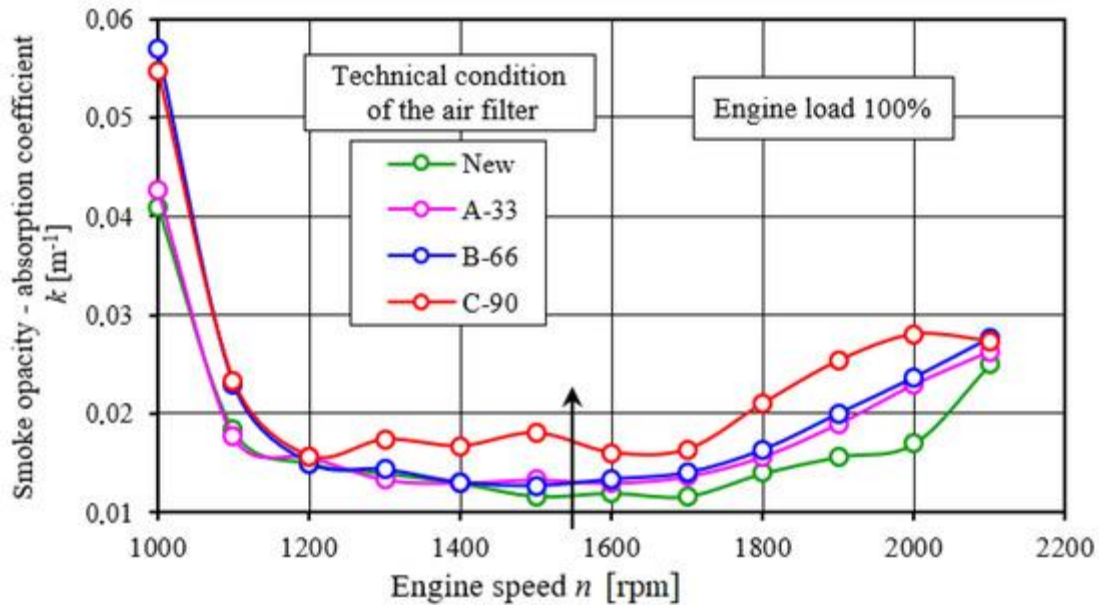


Figure 26. VOVLO DC13C460 engine smoke opacity—light absorption coefficient k (absorption) as a function of engine speed n for different air filter states: New, A-33, B-66, C-90.

[Figure 21](#) shows the pressure drop of four technical states (New, A-33, B-66, C-90) of the same air filter as a function of engine speed n of a Volvo D13C460 EURO V engine. The analysis of the test results was performed over the entire range of speeds that are used in the operation of a truck tractor equipped with the type of engine that was tested. As the engine speed increases in the range of $n = 1000\text{--}2100$ rpm, the pressure drop of the air filter, regardless of the percentage obscuration of the active area of the cartridge, increases its value until the engine speed reaches $n = 1900$ rpm, which is associated with the achievement of the maximum air demand by the engine ([Figure 20](#)) and the maximum power. A further increase in the engine speed causes a decrease in the filter pressure drop, which results from a decrease in the air flow rate Q_s . When 33% of the active surface of the cartridge (A-33) was obscured, no significant

differences in the pressure drop values were found when compared to the new filter (New). The recorded differences of the pressure drop are within the limits of the measurement errors and do not significantly affect the other parameters of engine operation presented in the following figures.

Obscuring 66% of the filter cartridge surface caused an increase in the filter pressure drop in the range of 10–15%, which at $n = 1900$ rpm reached a maximum value of 0.757 kPa. A significant (more than threefold) increase of the filter pressure drop in comparison to the filter with “New” cartridge was obtained only after covering about 90% of the cartridge active area. At $n = 1900$ rpm the filter reaches a maximum value of 2.024 kPa. For this air filter, the manufacturer has determined the permissible resistance of $\Delta p_f = 6$ kPa measured at $n = 1900$ rpm. A sensor installed in the duct after the air filter is used to record this value. Obtaining this value by the air filter means the necessity of performing maintenance consisting in filter cartridge replacement. The authors obscured 90% of the surface area and did not obtain an acceptable value, suggesting that only mineral dust retained on paper can provide an effective barrier to airflow. The low (0.604 kPa) value of the pressure drop of the new air filter (“New”) results from the large ($A_c = 13.72$ m²) filter area of the paper, which guarantees a low velocity of air flow through the paper surface, called in the literature the filtration velocity u_F calculated from the relation

$$u_F = Q_s / 3600 \cdot A_c \text{ [m/s]}, u_F = Q_s / 3600 \cdot A_c \text{ [m/s]},$$

(7)

where: Q_s —engine air demand [m³/h], A_c —filter cartridge active surface area [m²].

In the case of the tested engine, the air demand Q_s , irrespective of the technical condition of the air filter, increases rapidly until the engine reaches the rotational speed $n = 1900$ rpm, at which it reaches its maximum, and then it decreases ([Figure 21](#)). The use of an air filter with increasing pressure drop according to the variants (New, A-33, B-66, C-90) causes a shift of the characteristics $Q_s = f(n)$ almost in parallel towards lower values of the air demand Q_s by the engine. At speed $n = 1900$ rpm, the decrease in Q_s , is 1.15, 2.16, 3.39%.

For the maximum air demand $Q_s = 1387$ m³/h ([Figure 21](#)) and for the filter area $A_c = 13.72$ m², according to relation (7), the filtration velocity assumes the value $u_F = 0.0281$ m/s. This value is 50% lower than the permissible value of filtration velocity $u_{Fmax} = 0.06$ m/s, which is assumed during air filter design works. The low filtration velocity value guarantees a long period between air filter maintenance.

[Figure 22](#) shows the effect of four (New, A-33, B-66, C-90), differing in pressure drop, technical states of the air filter on the characteristics of the effective power $N_e = f(n)$ and specific fuel consumption $g_e = f(n)$ of the Volvo D13C460 EURO V engine. As the rotational speed increases, the effective engine power N_e , irrespective of the technical condition of the air filter, increases sharply in value until the engine reaches a rotational speed of $n = 1400$ rpm, and then decreases slightly until a rotational speed of $n = 1900$ rpm is reached, after which it loses its value sharply. The use of an air filter Δp_f with increasing pressure drop according to the technical states (A-33, B-66, C-90), shifts the characteristics $N_e = f(n)$ almost in parallel towards lower values of engine power. The decrease in power due to air filter contamination for a tractor-trailer type vehicle has a very negative effect on its traction characteristics. As the pressure drop increases and the power decreases, the traction properties of the vehicle deteriorate, in particular: the ability to climb a hill in individual gears, the change in the value of maximum speed depending on the road inclination angle and the time and distance of acceleration of the vehicle—tractor—trailer combination [\[74\]](#).

As the engine speed increases, the specific fuel consumption g_e , regardless of the technical condition of the air filter, increases slowly until the engine speed reaches $n = 1900$ rpm, and then increases significantly. In the case of obscuring 33% of the active area of the cartridge, no significant differences (more than 1%) were found in the change of the useful power N_e and specific fuel consumption g_e . Further increases of the air filter pressure drop (B-66, C-90) cause a significant effect on the decrease in power and a slight increase in specific fuel consumption—[Figure 22](#). At rotational speed $n = 1900$ rpm, the decrease in power due to increasing air filter pressure drop (A-33, B-66, C-90) takes the following values, respectively: 0.029, 2.31, 9.31%. At the same speed, the increase in specific fuel consumption takes the following values: 0.39, 1.74, 2.52%.

This phenomenon is due to the control strategy of the engine working process. In connection with the reduction of the boost pressure ([Figure 23](#)), the fuel dosage is changed in such a way as to increase the energy of the exhaust gases in order to increase the value of the intake air boost pressure to the required value, which is programmed in the controller map. However, changing the fuel metering control does not 100% compensate for the filling loss due to the increase in air pressure drop. In addition, the fuel metering control strategy only allows the fuel metering to be changed within a certain range. Significantly increasing

the basic fuel dose and delaying the injection angle leads to a significant increase in exhaust temperature and increased toxic emissions. In order to prevent such a situation and to ensure that the engine meets the Euro V standard, the controller operates appropriate algorithms in this regard.

At the rotational speed $n = 1600$ rpm, the decrease of the boost pressure p_d caused by the increase of the air filter pressure drop (A-33, B-66, C-90) over the value of the pressure drop of the new "New" takes the following values, respectively: 0.389, 1.95, 4.28%.

Figure 24 shows the hourly fuel consumption G_e as a function of engine speed n of a Volvo D13C460 EURO V engine for four air filters differing in pressure drop (New, A-33, B-66, C-90). As the engine speed increases, the hourly fuel consumption G_e , irrespective of the technical condition of the air filter, increases its value sharply until the engine speed $n = 1400$ rpm is reached. In the speed range $n = 1400$ –1900 rpm, the hourly fuel consumption G_e stabilizes at an almost constant level, after which it decreases sharply.

In the case of obscuring 33% of the active surface of the cartridge (A-33), no significant differences (about 0.75%) in the change of hourly G_e fuel consumption were found. Further increases of air filter pressure drop (B-66, C-90) already cause a significant decrease of hourly G_e fuel consumption, respectively: 3.21 and 7.87% (**Figure 24**). This should be explained by the fact that increasing the air filter pressure drop causes a decrease in the air mass delivered to the engine Q_s (**Figure 21**) and the boost pressure (**Figure 23**). A smaller mass of air delivered to the engine will not ensure accurate and complete combustion of fuel in the cylinders, resulting in excessive smoke and increased emissions of toxic exhaust components. In order to counteract this phenomenon, the ECU causes reduction of the maximum dose of fuel fed to the cylinders, which is reflected in reduction of hourly fuel consumption and power output. The reduction in hourly fuel consumption is due to the engine control strategy coded into the ECU, which is optimized to reduce emissions of toxic exhaust components.

Figure 25 and **Figure 26** show the exhaust gas temperature t_s and smoke k as a function of engine speed n of a Volvo D13C460 EURO V engine, for four air filters differing in pressure drop (New, A-33, B-66, C-90). As the engine speed increases, the exhaust temperature t_s , irrespective of the technical condition of the air filter, decreases slowly but systematically (almost linearly) its value until the engine speed $n = 1800$ rpm is reached, after which it increases slightly and then decreases sharply. Operation of the engine with an air filter Δp_f with increasing pressure drop, according to the technical states (A-33, B-66, C-90), causes a shift of the characteristics $t_s = f(n)$ almost in parallel towards lower values of the engine temperature. Technical states A-33, B-66 do not cause significant changes in exhaust gas temperature. Operation of the engine with an air filter with a pressure drop $\Delta p_f = 2.024$ kPa (C-90) already causes a significant, approximately 5%, decrease in the exhaust gas temperature t_s when compared to operation of the engine with the "New" filter. In the rotational speed range $n = 1000$ –1800 rpm the exhaust gas temperature t_s decreases by about $t_s = 20$ –30 °C, which is connected with smaller engine filling and decrease of the amount of fuel delivered to the cylinder in one working cycle of the engine.

The highest smoke opacity was recorded in the speed range $n = 1000$ –1100 rpm. However, as the engine speed increased, the smoke level decreased rapidly, irrespective of the technical condition of the air filter, and remained stable in the speed range $n = 1100$ –1700 rpm, after which it increased slightly. However, the increase in air filter pressure drop does not cause significant changes in the degree of smoke opacity in relation to its permissible value, defined as for the technical conditions of vehicle operation for this type of vehicles at 1.5 m^{-1} [75].

4. Conclusions

1. In the available literature there are no results of investigations of the influence of the flow resistance in the inlet system on the performance of the modern diesel engine used for driving of the trucks and truck tractors currently travelling on the roads and constituting the basic means of transport of goods.

2. The basic filtering material for motor vehicle internal combustion engine operating fluids are fibrous materials, including filter paper, which provides high efficiency (above 99.5%) and accuracy (above 5 μm) of filtration. The disadvantage of filter papers is low dust absorption (220–250 g/m^2), which limits the lifetime of the filter.

3. During the use of baffle air filters for inlet air of vehicle engines, due to the retention of dust grains, the air filter pressure drop increases systematically. For this reason, after the mileage of the vehicle specified by the manufacturer or when the filter reaches a certain value of the permissible resistance Δp_{fdop} , the filter insert must be replaced.

4. It is generally assumed in the construction of internal combustion engines that the air filter may have such a maximum pressure drop that at the maximum power speed of the engine and its full load (100% opening of the mixture throttle in the case of engines with SI and full fuel dose in the case of diesel engines) does not cause a greater drop in power than 3%, rarely 5%. This condition limits the increase of pressure drop resulting from its dust loading during operation. The value of the permissible pressure drop Δp_{flop} shall be selected individually for each engine depending on the demand for air and the anticipated operating conditions, and in particular on the concentration of dust in the air.
5. The use of air filter after exceeding the value of admissible resistance Δp_{flop} (not replacing the filter cartridge) is possible as the filter still retains high filtration efficiency. However, there is a sharp increase in the pressure drop, which causes additional loss of engine power and deterioration of the vehicle's traction properties, in particular its climbing ability.
6. From the analysis of the available test results of the engines found in the literature, it results that the increase in the resistance to the flow of the air filter Δp_f by 1 kPa causes on average a decrease in the power of the SI engine (carburetor) of (1.0–1.5)% and an increase in specific fuel consumption of about 0.7. For diesel engines with a classical injection system, these values are respectively (0.4–0.6)% decrease of power and (0.3–0.5)% increase of specific fuel consumption.
7. The research carried out on an engine dynamometer on the influence of the air filter pressure drop on the parameters of the modern, supercharged, with an inlet air cooler diesel engine, being a driving unit of a truck tractor, only partly confirms the previous research results presented in the literature. This is due to different injection systems and fuel delivery strategies, which in modern engines is aimed at minimizing smoke and reducing emissions of toxic exhaust components, rather than maximizing power.
8. The increase of 2 kPa in the air filter pressure drop above that of the “New” one results in a 9.31% decrease in the effective power of the tested engine, which is 4.66% per 1 kPa increase in the pressure drop and is 10 times higher than in the case of diesel engines with classic injection systems. The decrease in power is correlated with a decrease in engine inlet flow (3.39%) and boost pressure (4.28%), a decrease in hourly fuel consumption (7.87%) and an increase in specific fuel consumption (2.52%).
9. The decrease of the engine power caused by the increase of the flow resistance in the intake system (air filter contamination) will have a very negative effect on the traction properties of the vehicle (truck tractor), from which the tested engine originated. As the engine power decreases, the traction properties of the vehicle deteriorate, in particular: the ability to overcome a hill in individual gears, the change of the maximum speed value depending on the road inclination angle and the time and distance of acceleration of the vehicle–tractor–trailer combination.
10. Increase of the flow resistance in the inlet system of the tested engine did not significantly affect the increase of smoke, which is a very beneficial phenomenon, since there is no increase in the emission of particulate matter, which is currently one of the most significant components polluting the atmosphere.
11. Comparing the test results obtained with the results presented in the literature for older engine types, it should be stated that the modern truck engines with electronically controlled fuel injection and turbocharging systems are less susceptible to the negative influence of contamination of the intake system in comparison with engines with mechanical supply systems— injection pump, carburetor. This is due to the fact that the ECU of the modern engine on the basis of input signals—boost pressure, mass of intake air, vacuum in the intake system—corrects the fuel dose in such a way as to offset the negative effects of increased resistance to flow. The dose of fuel and the method of its application is selected in such a way as to limit power loss with no increase in smoke.
12. The obtained results showed the influence of the flow resistance in the inlet system of the modern truck engine on its performance. It is advisable to continue the work, the final effect of which should be the determination of the maximum/admissible flow resistance, the exceeding of which should eliminate the vehicle from further operation due to deterioration of the economic, energetic, ecological and traction properties of the vehicle.
13. It is advisable to extend the presented research by an analysis of the influence of increased pressure drop in the air filtration system on the emission of toxic components of exhaust gases and a change in the traction characteristics of the vehicle equipped with this type of inlet system.

Author Contributions

Conceptualization, T.D. and M.K.; methodology, M.K.; software, T.D.; validation, T.D. and M.K.; formal analysis, T.D. and M.K.; investigation, T.D. and M.K.; data curation, T.D. and M.K.; writing—original draft preparation, T.D. and M.K.; writing—review and editing, T.D.; visualization, T.D. and M.K.;

supervision, T.D. and M.K.; project administration, M.K.; funding acquisition, T.D. and M.K. All authors have read and agreed to the published version of the manuscript.

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Data Availability Statement

Data is contained within the article.

Conflicts of Interest

The authors declare no conflict of interest.

Abbreviations

ε	compression ratio
ζ	dimensionless pressure drop coefficient of the inlet system
G_e	hourly fuel consumption
g_e	specific fuel consumption
i	number of cylinders
k	smoke opacity of exhaust gases—light absorption coefficient
κ	stroke number factor
L_t	theoretical air demand
λ	excess air factor
M_o	engine torque
N_e	effective engine power
$\dot{m}_{rz}$	mean real flow rate of air supplied to engine cylinders
$\dot{m}_t$	theoretical mass flow rate of air supplied to engine cylinders
n	engine rotational speed
η_i	indexed efficiency
η_m	mechanical efficiency
η_v	filling factor

p_d	charge air pressure
p_H	ambient pressure
p_N	pressure at the end of the filling stroke
p_r	pressure of the rest of the exhaust gas
p_1	pressure before
Δp_f	air filter pressure drop
T_H	ambient temperature
T_N	temperature at the end of the filling stroke
T_r	temperature of the rest of the exhaust gas
Q_s	engine air demand
V_s	cylinder displacement
v_L	average charge velocity in the inlet system
W_d	fuel calorific value
ρ_{pow}	air density
ρ_L	charge density

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