MONITORING OF STATE QUANTITIES OF DIESEL ENGINES IN LABORATORY CONDITIONS

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Worsening of ecological parameters of an engine causes a pressure drop in the combustion chamber or declines the technical condition of the fuel system (Hofmann, Rauscher, 1996). The present experiment describes the possibility of low pressure turbocharging at the atmospheric diesel engine and provides necessary information on the state quantity pressure at the end of the compression stroke. Contribution points out the effect of wear of functional pairs in combustion chamber and a subsequent decrease of the level of air internal energy at the end of the compression stroke. The observed diesel engine is atmospheric, so its low pressure supercharging can be ensured from a separate air tank. Simulation of the worn engine was secured by changing the exhaust valve's clearance to 0.50 mm for comparison of the measured values, so the limit wear state reached 28%. The objectives of the experiment were fulfilled, and applied low-pressure supercharging has been proved. On the basis of the measured data, the detected value of the supercharging was just 40 kPa, while the pressure at the end of compression stroke of the simulated worn engine equaled to the pressure of the new engine at the selected speed.

emission; diesel combustion; pressure drop; turbocharging

INTRODUCTION

The use of diesel engines within the EU represents over 50%, which is a big increase if compared with the end of the last century. In accord with the worldwide trend, the manufacturers of combustion engines are subjected to various requirements and criteria. These criteria include technological, economic, and ecological aspects with great emphasis. Diesel engines manufacturers are constantly trying to develop and construct these engines to be as economic and as powerful as possible. At present, modern diesel engines are naturally expected to run a high number of kilometers without loss of performance and possibly without increased oil consumption (Shayler et al., 2002).

After a certain time, factors acting during the life span of the diesel engine may have a negative impact on its functioning. These factors can be roughly divided into so called "fuel" and "atmospheric". Atmospheric factors are atmospheric disturbances which are constantly changing and depend on the weather situation. The atmospheric disturbances which mostly influence the quality of the suctioned air are the atmospheric pressure (altitude), humidity, and air pollution. Air pollution is in direct connection with the quality of the suctioned air and causes clogging of the engine parts (Ilkilic, Aydin, 2011). Air temperature largely influences the fact whether the fuel burns completely. Suction at low ambient temperature can significantly influence the actual performance of the engine which can be called as cold suction. Generally we may say that every temperature increase of the suctioned air by 4 °C reduces the performance by 1% (Urban, Rusnák, 1998).

The technical condition of a diesel engine directly affects its performance and economic parameters. Reliability of diesel engines can be estimated by different methods which are given by their size, installation, and available equipment (Halderman et al., 2004). The most common method of diesel engines estimation is technical diagnostics which utilizes measurement methods resulting from specific requirements of this diagnostics (Vlk, 2006). Basic information can be obtained by direct measurement of characteristic quantities by suitable equipment and instruments (Bergmann et al., 2009).

MATERIAL AND METHODS

Deterioration of components in the combustion chamber gradually changes the run of the engine which leads to its ultimate state of operation (Jakubík et al., 1998). This deterioration also results in reduc-

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tion of the internal energy of the air and worsens the economic and ecological parameters of the engine(Greuter, Zima, 2000).

In the experiment we deal with a possibility of attaining such internal energy of the air at partially worn engine so that we could approach the internal energy of a new engine. The solution could be in applying low-pressure turbocharging. The partially worn engine has been simulated by alteration of the exhaust valve gap what caused an incomplete contact of the valve with the cylinder head saddle. The required exhaust valve gap was made by an indicating gage supported against the non-operating surface of the tripper device (Fig. 1). The adjustment of the valve gap was set to 0.50 mm and thus we reached the state change at the end of the compression stroke corresponding to the wear of 28%.

The measurements were conducted on four-stroke four-cylinder diesel engine Zetor 8401.12. This engine is not turbocharged with direct injection of fuel. Its cubature is 4562 cm^3, theoretical compression ratio is 17 and maximal combustion pressure at given revolutions is 8.3 MPa.

The above mentioned low-pressure turbocharging is ensured by charging the cylinder from a special air reservoir whose volume is 220 times greater than the volume of the measured cylinder. Connection between the reservoir and the cylinder is realized by a specific pipe with an internal diameter of 52.2 mm. In this case the supply pipe leads from the reservoir in the axis of the suction channel of the cylinder. Prior to cylinder flanges mounting is the displacement of the complete suction and exhaust system by one cylinder forward (Fig. 2).

The supply line is equipped with two separate valves. Valve 1 (11) in first position ensures suction of atmospheric air and in second position stops this suction. Valve 2 (12) is placed in-between the measured cylinder and the reservoir and in first position it ensures stopping of charging pressure from the reservoir and in second position it stops charging of the cylinder from the air reservoir. In order to monitor the pressure in the reservoir and also in the charging pipe, there are two separate pressure gauges (9 and 10).

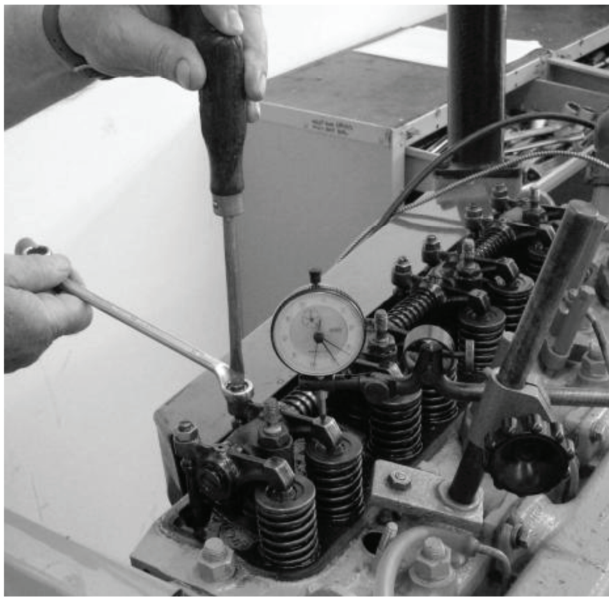

Fig. 1. Adjustment of the exhaust valve

The main state quantity monitored during the experiment is the pressure at the end of the compression stroke. Measurement of the pressure at the end of the compression stroke is carried out by a dynamic method using piezoelectric sensor. Piezoelectric sensor is placed inside the combustion chamber of the measured cylinder by means of a special adaptor (Fig. 3).

The presence of the adaptor is important in order to keep the compression ratio of the cylinder.

The state quantity was measured by a diagnostic device Kistler 2507 A. This device is designed for dynamic measurement of compression pressure of diesel

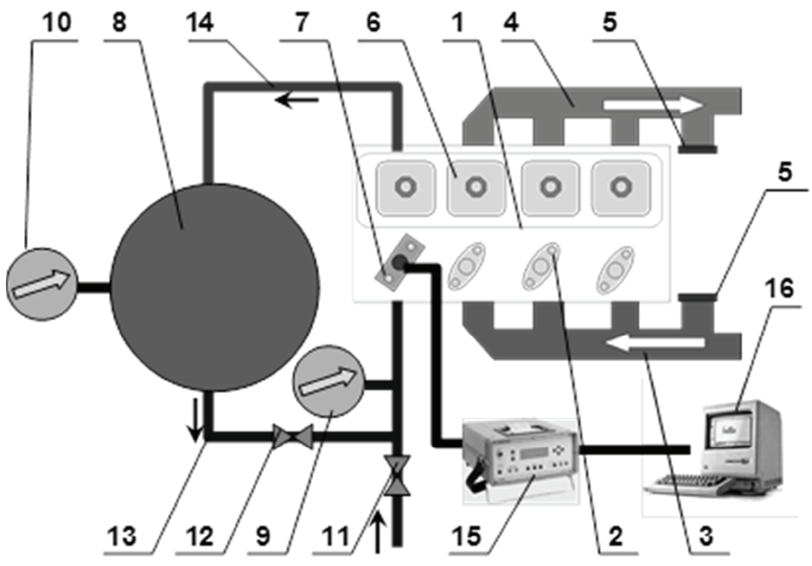

Fig. 2. Connection of charging equipment and measurement chain
1 = cylinder head, 2 = injector, 3 = suction pipe, 4 = exhaust pipe, 5 = tightening flange, 6 = measured cylinder, 7 = piezoelectric sensor with adaptor, 8 = charging reservoir, 9 = pressure gauge of charging pipe, 10 = reservoir pressure gauge, 11 = two-position valve of suction, 12 = two-position valve of charging, 13 = charging pipe, 14 = supply line, 15 = measurement and evaluation unit kistler 2507a, 16 = software controlled computer for data processing

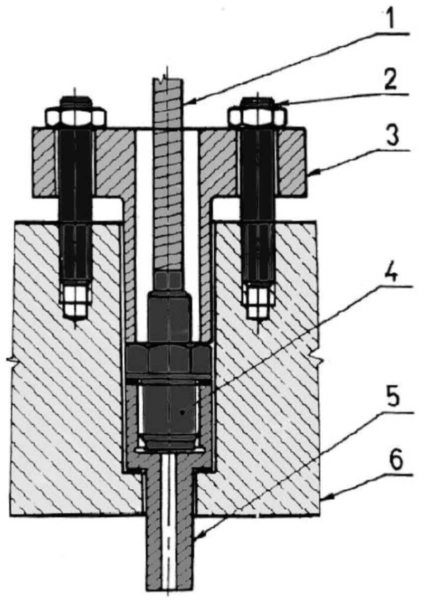

Fig. 3. Built-in adaptor with piezoelectric sensor
1 = communication cable of the sensor, 2 = tightening matrices,
3 = sensor case, 4 = piezoelectric sensor, 5 = sensor adaptor,
6 = cylinder head

engines. It measures a non-electric quantity, pressure at the end of the compression stroke, revolutions and electric quantity converted by a piezoelectric sensor Serridge, Licht, 1986). The built-in electronics is controlled by a program where 40 consecutive pressure cycles are automatically included, from which the arithmetic mean value and dispersion is calculated.

Upon connection of the sensor and the diagnostic device to the logic measuring chain (Fig. 2), the preparation of the engine for measurement follows. The engine preparation consists of the oil level checking, the adjustment of the valve gap, cleaning of the air filter as well as of checking of the adaptor fixation and of the installation of auxiliary pipe for the fuel drainage. Prior to running of the engine it is necessary to provide forced cooling of the engine. The last step of the preparation is heating of the engine to the temperature of 90°C. The methodology of the measurement is completed by the information that the engine was not loaded during the experiment.

RESULTS AND DISCUSSION

According to Hatschbach, Baumruk (2002), knowledge of the internal aerodynamics of the diesel piston combustion engine is very important for further improvement of its performance, environmental parameters, and customer requirements as well as for meeting the strict legislative standards. The monitored measured object Z-8401.12 engine is equipped with a direct fuel injection with a toroidal combustion chamber in the piston. Engines with the direct injection should have stable layers of mixture and low between-cyclic variations which make it possible to stabilize a current field especially in the top dead centre.

By performing of all conditions given in the methodology of measurement it is possible to start the measurement itself (Hrubec et al., 2002). The obtained data are listed in Table 1. The obtained data on the pressure at the end of the compression stroke were evaluated by the built-in digital sensor which processes the mean value and the sphere of dispersion through more than 40 impulses. From these data only the peak value of the pressure is presented.

Created characteristics of the pressure at the end of the compression stroke in dependence on charging pressure with raising revolutions can be seen in Fig. 4. The revolutions of the engine were gradually increased from 800 to 2000 min^{-1} in intervals of 200 min^{-1} and actually measured revolutions are depicted in Fig. 4. In the right part of this figure we can see charging pressures, together with colour distinction of the curves, which were increased by 10 kPa (from 0 to 80 kPa). This graphical display represents the increasing character of respective curves, hence the pressures between measurements increase in regular intervals. For better illustration we have included the characteristics with zero charging pressure.

Comparison of measurement series at the simulation of cylinder charging as well as the pressure increase is shown in Fig. 5. From this figure it is possible to read the lowest difference in pressure increase as 0.32 MPa at revolutions 800 min^{-1} with charging 10 kPa and the greatest difference in pressure increase as 0.44 MPa with charging 40 kPa. At the highest preset revolution of 2000 min^{-1} it is possible to see the lowest difference in pressure increase as 0.28 MPa with charging 10 kPa and the greatest difference in pressure increase as 0.56 MPa with charging 80 kPa.

Table 2 shows the measured values of pressure at the end of the compression stroke at charging up to 40 kPa on partially worn engine. The following characteristics (Fig. 6) compare the pressures at the end of the compression stroke obtained from two regimes of measurement, namely on a "new" and "partially worn" engine (distinguished by colour in legend). The given characteristics represent a gradual pressure increase in complete range of revolutions. The greatest pressure difference at the end of the compression stroke between the "new" and "partially worn" engine is 1.07 MPa at revolutions 800 min^{-1}. The lowest difference in pressure is 0.85 MPa at revolutions 2000 min^{-1}. These results demonstrate the fact that by charging of the cylinder by low-pressure turbocharging with the pipe diameter of 52.2 mm we reached the pressure at the end of the compression stroke at worn engine compared to a new one at the value of the charging 40 kPa in a thorough range of revolutions.

Table 1. Pressure at the end of the compression stroke in dependence on charging pressure

Charging pressure p_p (kPa)	10						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	815	1013	1218	1421	1609	1813	2033
Pressure at the end of compression p_K (MPa)	4.12	4.2	4.34	4.52	4.62	4.82	4.86
Charging pressure p_p (kPa)	20						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	811	1014	1217	1419	1606	1814	2016
Pressure at the end of compression p_K (MPa)	4.5	4.56	4.68	4.9	5.06	5.24	5.32
Charging pressure p_p (kPa)	30						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	810	1013	1216	1418	1604	1812	2015
Pressure at the end of compression p_K (MPa)	4.84	5	5.12	5.32	5.52	5.64	5.76
Charging pressure p_p (kPa)	40						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	810	1012	1217	1418	1603	1810	2016
Pressure at the end of compression p_K (MPa)	5.28	5.34	5.54	5.74	5.96	6.16	6.24
Charging pressure p_p (kPa)	50						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	807	1013	1216	1419	1602	1809	2018
Pressure at the end of compression p_K (MPa)	5.66	5.76	5.98	6.2	6.42	6.62	6.72
Charging pressure p_p (kPa)	60						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	806	1011	1215	1418	1600	1806	2013
Pressure at the end of compression p_K (MPa)	6.06	6.22	6.38	6.6	6.76	6.96	7.18
Charging pressure p_p (kPa)	70						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	805	1013	1214	1417	1600	1805	2014
Pressure at the end of compression p_K (MPa)	6.42	6.62	6.82	7.06	7.26	7.5	7.6
Charging pressure p_p (kPa)	80						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min–^{1})	806	1001	1215	1417	1600	1802	2012
Pressure at the end of compression p_K (MPa)	6.78	7.1	7.22	7.48	7.68	7.98	8.16

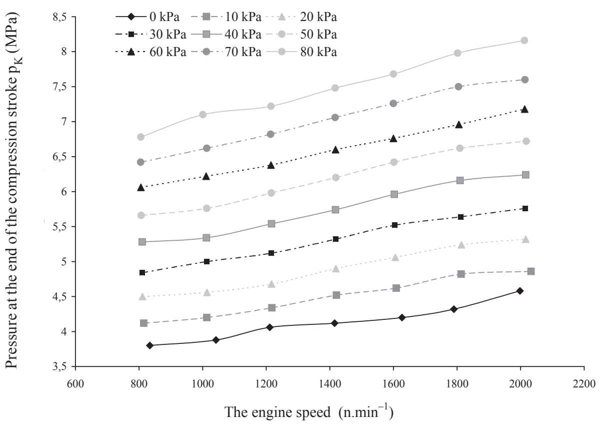

Fig. 4. Characteristics of the pressure at the end of the compression stroke of the Z-8401 engine in dependence on charging pressure

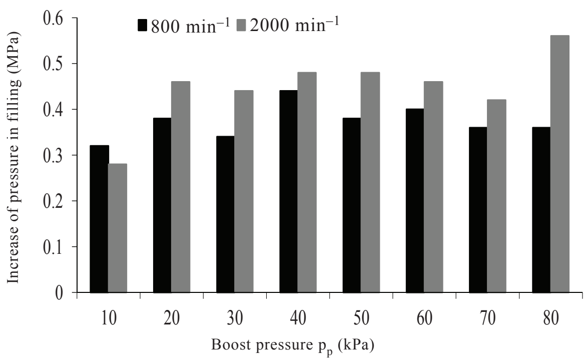

Fig. 5. Pressure increase at charging as a function of revolutions

Table 2. Pressure at the end of the compression stroke in dependence on charging pressure of the worn engine

Charging pressure p_p (kPa)	10						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	811	1029	1217	1418	1610	1807	2014
Pressure at the end of compression p_K (MPa)	2.6	2.96	3.18	3.08	3.52	3.78	4.04
Charging pressure p_p (kPa)	20						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	812	1026	1211	1422	1610	1807	2012
Pressure at the end of compression p_K (MPa)	2.92	3.3	3.5	3.52	3.78	4.16	4.46
Charging pressure p_p (kPa)	30						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	809	1024	1211	1421	1609	1808	2012
Pressure at the end of compression p_K (MPa)	3.22	3.64	3.86	3.88	4.18	4.54	4.86
Charging pressure p_p (kPa)	40						
Set revolutions n_{zv} (min^{-1})	800	1000	1200	1400	1600	1800	2000
Actual revolutions n_m (min^{-1})	813	1021	1209	1416	1609	1808	2015
Pressure at the end of compression p_K (MPa)	3.58	3.94	4.2	4.22	4.52	4.92	5.28

CONCLUSION

The main objective of the experiment was the possibility to use low pressure turbocharging by means of simulation at a selected atmospheric diesel engine and the comparison of obtained information on the state quantity at "new" and "partly worn" engine. Charging of the given cylinder was performed by an air reservoir. The cubature of the given cylinder was 1140.5 cm^3 and the cubature of the reservoir was 220 times greater, i.e. 250 910 cm^3.

The information obtained from pressure characteristics at the end of the compression stroke in dependence on charging pressure (Fig. 4) provide a comprehensive image of gradual pressure increase and cylinder charging. Dependence with zero charging pressure has been included in order to simplify the comparison with a non-turbocharged diesel engine. All variables have an increasing character which meets the theory of pressure increase at the end of the compression stroke depending on increasing revolutions.

From the comparison of pressure increase at preset revolutions between respective charging (Fig. 5) it is possible to read unsteadily distributed values. At revolutions 800 min^{-1} the pressure difference is 0.32 MPa between two different measured states. This indicates that at charging 10 kPa it is possible to increase the state variable by 0.32 MPa. At maximal preset revolutions 2000 min^{-1} with charging 10 kPa we reached by a 0.28 MPa lower increase of the state variable. All measurements have been carried out in order to verify low-pressure turbocharging on a partially worn engine and to find the parameter of charging that approaches the characteristics of a non-worn engine at the end of the compression stroke.

The characteristics given in Fig. 6 clearly demonstrate the suitability of low pressure turbocharging at the atmospheric diesel engine in the whole range of revolutions even at 40 kPa of the charging pressure. The obtained results indicate the advantages of the used turbocharging with respect to the simple construction of the device without the necessity of external

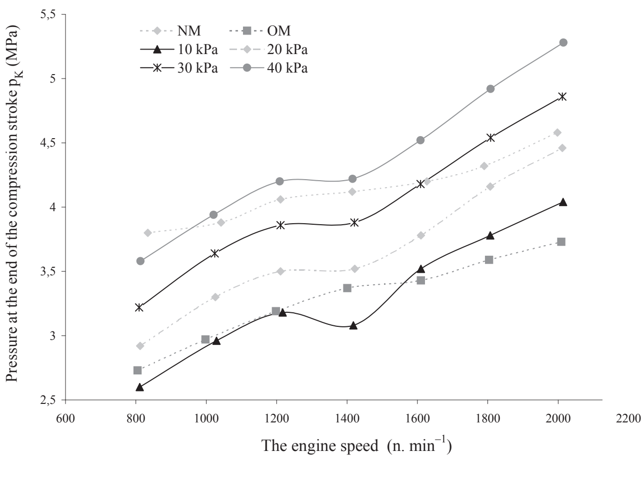

Fig. 6. Comparison of pressure at the end of the compression stroke on a "new" (NM) and "worn" (OM) engine in dependence on charging pressure

source of compressed air. The proposed device for low pressure turbocharging is suitable also from the economical viewpoint since it does not require high financial investment into necessary accessories and spare parts.

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