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DEVELOPING ENGINE TECHNOLOGY

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Preface

Since 19th century, or maybe even earlier, i.e. the time of the founders of combustion engine theories and the engine constructors, their incessant development continues.
Today, nobody thinks about why or how an engine works. The constructors focus their attention on wide-ranging issues of efficiency of this heat machine, in terms of ecology and economical operation. This is so because, despite numerous parallel studies on electric propulsion or fuel cells, combustion engines still remain the most popular source of drive, and it will probably continue for another 30 or more years.
One can find an innovative idea in any system or even element of an engine. These include modern fuel injectors, which allow dividing a fuel dose into several parts during one operation cycle. Construction solutions concerning variable valve timing or valve lift are still being modified. Modern computer-aided systems eliminate the unbalanced primary and secondary forces. Introduction of successive changes in the supercharging systems, just as the whole process referred to as downsizing, are only some evidence of development works on combustion engines completed in recent years. It is difficult not to mention the cooperation among classic mechanics, chemists and electronic specialists, forming interdisciplinary research teams. It is thanks to them that we learn more about alternative fuels or the methods of microprocessor control indispensable in achieving the current legislative guidelines concerning combustion engines.
In this endless recital of achievements or development work concerning combustion engine, one thing remains constant – the need for verification or diagnostics of introduced solutions through lab research, and then in natural operation. Naturally, there is also considerable improvement in the area of measurement techniques, but the idea remains unchanged – determination of the combustion engine characteristics.
Given the vast diversification of the existing formulas defining the relations between engine operating parameters, its fullest image of operation is obtained by way of the engine performance map. It expresses the relation of various performance to engine speed and its load, defined by torque or mean effective pressure. The most popular map is the image of a constant specific fuel consumption and the hyperbolas of constant power. It depends on the adopted research target what other parameter has to be determined. In this way the engine performance maps can express the relation of speed and load, to e.g. concentration of toxic components of exhaust gas or the temperature in the deposit of a catalytic converter.

This work is designed for those who manage or will deal with combustion engine research. The book provides basic information on research organization, test stand, research methods and measurement errors and the ways of their estimation. The successive chapters present a review of basic parameters of engine operation, and description of the ways they are determined. Thermal balance items were also discussed along with a presentation on how to create various characteristics, and special attention was paid to the performance maps.
Numerous images, graphs and tables included here should be helpful in getting familiar with the material (and make it more complete), and those more inquisitive ones can take advantage of the literature references.
This book is not intended as the only and correct compilation of the material included in it. The authors hope it will be helpful in teaching specialists in Automotive Engineering.

Authors

Contents

ABBREVIATIONS AND SYMBOLS

A_t - surface area of the piston crown,
B.s.u. - Bosch scale units
BDC - Bottom Dead Centre
bmep - brake mean effective pressure
B_o - fuel charge per cycle, kg
bsfc - brake specific fuel consumption
Ca - constant, depending on the design parameters of the screw propeller.
CAD - crank angle degree
CLD - chemiluminescence analyser
C - orifice constant
c_w - specific heat of the cooling liquid
d_1 - orifice diameter
D - diameter of the inlet channel
delta p/delta α - speed of a pressure increase.
δ_y – relative error
e_M - torque flexibility ratio
e_n - engine rotational speed flexibility ratio
e - total flexibility of an engine
$f(x_i)$ - regression function (unknown),
f_a- factor taking account of atmospheric conditions,
F - force applied to dynamometer arm of length equal to „b";
FID - flame ionizing detector
f_m - factor taking account of the engine type and regulation
FSN - filter smoke number
GC - Gas Chromatograph
Ge - fuel consumption per hour (fuel flow)
g_e - specific fuel consumption
i - number of cylinders
K - brake constant,
k - isentropic exponent of the tested gas
Ko - correction factor to normal conditions
L_e - effective work
m - cooling medium streams
Mc_p - molar specific heat at constant pressure,
Mc_v - molar specific heat at constant volume
M_{omax} - maximum torque
M_{oN} - torque of maximum useful power of engine
Mo - torque
MPD - paramagnetic analyser for determining oxygen concentration
m_p - mass flow rate (intensity)
MR - universal gas constant (MR = 8314,3 J/(kmol K),
NDIR - nondispersive infrared
n – engine's rotational speed

n_g - rotational speed at which g_e, reaches its minimum value.
n_{max} - maximum rotational speed at which the engine can operate without any disturbance.
n_M – engine's rotational speed at maximum torque
n_{min} - minimum amount of oxygen needed for complete and perfect combustion of 1 kg fuel, kmol.kg
n_N - engine rotational speed at maximum useful power
n - number of kilomoles of gas per a unit of time,
n - number of results:
n_o - lowest rotation speed at which the engine can operate with the external load, overcoming the resistance to motion and internal friction,
n_p - number of kilomoles of sucked air,
n_{rz} - amount of air involved in the actual process of combustion
n_s - number of kilomoles of exhaust gas.
n_{teo} - amount of air necessary for full and complete combustion of fuel
p_{max} - peak pressure during combustion,
p_1 - absolute air pressure at the orifice
p - actual pressure of combustion
p_{comp_max} - peak compression pressure,
p_e - brake mean effective pressure
Pe - useful power (brake power) of engine
P_i - indicated power
p_i - mean indicated pressure
p'_{max} - value of peak pressure among recorded cycles,
p_{max} - values of peak pressure of single cycles,
p'_{minv} - value of minimum pressure among recorded cycles,
p_{ot} - ambient pressure, Pa
ppb - parts per billion
ppm - parts per million
p_T - average friction pressure
Q_{ch1} - thermal flux losses attributable to cooling of the cylinder block
Q_{ch2} - thermal flux losses attributable to cooling of the oil crankcase
Q_{ch} - heat carried off to the cooling medium,
Q_{ch} - heat losses attributable to cooling,
q_{ch} - share of cooling losses,
Q_{dys} - dissociation heat losses,
Q_e - effective thermal flux,
Q_{HV} - fuel calorific value,
Q_m - thermal flux attributable to resistance to motion,
Q_{ns} - the amount of heat lost due to incomplete fuel combustion,
Q_o - the total amount of heat supplied to the engine,
Q_p - thermal flux losses attributable to radiation of the cylinder block.
Q_r - the rest of the balance, containing indefinable radiation heat losses,
Q_s - heat

Q_w - exhaust heat losses,
q_r - share of the remaining losses.
q_w - share of exhaust losses,
R - dynamometer's indication,
S_{ny} - mean square deviation for a series of measurements
S - piston stroke,
S_y - mean square deviation for a single measurement
σ_y - standard deviation for a normal distribution
T - temperature
TDC - Top Dead Centre
T - measured consumption time of a controlled volume of fuel
T_{ot} - ambient temperature, K
t_x - fuel dose consumption time
V_{air} - volume of sucked air
V_{air} - volumetric flow rate
V_p - controlled volume of fuel consumed in time „t",
V_s - swept volume of one engine cylinder,
v_x - fuel dose volume,
W_e - useful work
W_i - indicated work
W_{il} - indicated work per cycle
W_t - theoretical work
x_i - input value (input function),
y_i - output value (measured),
y_{sr} - mean value
z" - shares of respective components,
α - flow number,
α^o_w - injection/ignition advance angle,
α^o_z - ignition delay angle,
ΔT - difference in the temperatures between the inflow and outflow of the cooling system
ε - expansion number,
η_c - thermal efficiency
η_e - effective (useful) efficiency
η_i - indicated efficiency
η_m - mechanical efficiency
η_m - mechanical efficiency
η_t - ideal efficiency
η_v - volumetric efficiency of a single cylinder
λ - excess air number
μ - molar enthalpy change factor (μ = 1.02 ¸ 1.04)
ν_l - specific volume of air in normal conditions
ζ - heat release rate
ρ_{fuel} - fuel density,

τ - number of working strokes
$\varphi = p_{max}/p_{comp\_max}$ - degree of a pressure increase,
ω - angular velocity of crankshaft (frequency of rotation)
%vol- volume percentage
Δp - differential pressure
Δyi - absolute measuring error

1. BASIC INFORMATION ON TESTING COMBUSTION ENGINES

1.1. Research procedure – general information

A **research** is an activity involving observation and experiment, which aim at gathering information on the studied phenomenon or subject. An **observation** is the perception of a given task in its natural setting of operation, whereas an **experiment** should be referred to as each activity leading to a deliberate triggering of some defined changes in the analysed task, with concurrent recording of such changes, measuring characteristic values and determination of the occurrence of research dependencies. A decided majority of studies are experiments whose scope and methods of execution are subordinated to their aims. This is why three types of experiments can be distinguished [7, 27]:

- diagnostic experiment, aiming at finding out features (properties) of a researched phenomenon or subject, classifying them according to adopted criteria, and generally speaking determining their state – good/bad,
- decisive experiment, which aims at deciding on selection of a hypothesis or theory for the needs of describing the task under consideration,
- binding experiment, aiming at finding out and defining quantitative or qualitative dependencies between parameters of the studied task.

Each of the above-mentioned experiments is carried out in compliance with a strictly defined research procedure, comprising the following stages:

- planning a research programme,
- an experiment design,
- carrying out research,
- analysing the results,
- recording the experiment results (in the latter period equated with the analysis).

The programme of the research is a document defining the procedure of an experiment. It includes, among others, the issue of formulating a problem, which entails determining an aim and pointing out an object of the research. In turn, the determination of the object of the research necessitates defining the values to be examined (temperature, pressure, speed etc.) and assuming their specific nature (slow, quick and constant values, etc.). When defining the tested value one needs to determine the accuracy with which it should be measured.

The formulation of the aim also necessitates establishing assumptions determining the conditions for running tests, as well as determining their methodology. A crucial element of a research methodology is to determine the ways by which such aim is to be achieved, i.e. a measuring technique, as well as specifying measuring devices or systems. In this case it is vital to ensure availability and financial resources of the person carrying out the research – colloquially referred to as an observer or experimenter.

Determination of the steps needed to complete the research, determination of their sequence and the conditions for conducting an experiment, in order to obtain the information on the research object with the required accuracy – they all form the procedure called the experiment design, which uses elements of mathematical statistics for creating an optimal research process [7, 27, 29].

Execution of the research should follow an adopted programme and a designed mathematical plan of the experiment. The research execution stage includes: designing and preparing a test stand, installation of measuring systems, measurement taking and collecting data. At this stage, the experiment can be modified, and this need may arise directly from the obtained results. Here, it must be remembered that any departure/deviation from the execution of the research must be preceded by modification of the assumptions, before the experiment is continued. In this way the experiment remains under control of the experimenter at all times.

The data collected during the research are processed in the result analysis. **The research results analysis** mainly involves a systematic approach to the collected data, usually performed using a relevant mathematical method. This analysis should result in accurate conclusions determining the appropriateness of adopted assumptions and the forms of execution, as well as the information about the completion of the research aim.

The last stage of the research procedure is the **recording of the research results**. Using automatic forms of experiment conduct, this stage of the research procedure takes place already during recording or analysis of results, which is why in some procedures it does not function separately.

The conducted analysis leads to a diagram of the research procedure, presented below.

A. RESEARCH PROGRAMME

A1.establishing the problem,

A2 formulating the aim,

A3 determination of the object

- level of the object's complexity (a whole, system, element, process, etc.),
- values under investigation (temperature, speed, force, etc.),
- specific features of values (constant, variable in time, etc.),
- measurement accuracy,

A4 establishment of assumptions

- conditions for completion of the research,
- limitations,
- interference,

A5 methodology of research

- standardization of research,
- measurement techniques,
- measurement devices and systems,
- automation of the experiment (data collection, control and monitoring),
- calibration,
- availability of instruments,
- financial capabilities

B. MATHEMATICAL EXPERIMENT DESIGN

C. COMPLETION OF RESEARCH

C1. design of the test stand,

C2 construction of the test stand,

C3 installation of measuring systems,

C4 generating of input functions,

C5 data collection.

D. ANALYSIS OF RESULTS

D1 systematic approach to the collected data,

D2 drawing conclusions,

D3 appropriateness of adopted assumptions,

D4 forms of execution (completion),

D5 achievement of the aim.

E. RECORDING OF RESULTS

E1 reporting (written description, tables and diagrams),

E2 recording the information on data carriers.

This research procedure is an open one, i.e. it can be extended with some details resulting from any needs of the experimenter, while the basic structure should remain unchanged. In the case of typical and frequent measurement tasks, the research procedure can undergo a standardization process which will ensure comparability of the execution conditions and the ways conclusions are drawn among various experiments.

1.2. Testing of combustion engines

The testing of a combustion engine represents the basic link in the process of its design. The co-existence in a combustion engine of chemical, mechanical and thermal phenomena, as well as the dynamics and periodicity of their occurrence, complex geometrical and functional composition of elements, and a vast diversity of types and applications legitimizes phrasing it a specific nature of the machine's research.

There are basically two types of experiments on combustion engines. These include an examination in natural operation and tests conducted at the test stand, the so-called engine test house. Due to a relatively easy provision of stable research conditions, resulting in drawing appropriate conclusions, the test stand research prevails over the operation testing in all three types of the experiment, i.e. diagnostic, decisive and binding experiments.

The research procedure of a combustion engine is compliant with the generally adopted one (see section 1.1.), but it is very individual in its approach (specific) at its each separate stage. The research process begins by establishing the **research programme**, and this in turn requires the experimenter to pose a problem and formulate an aim. The coincidence of various phenomena mentioned in the introduction to this chapter makes the scope of research problems relating to this thermodynamic machine vast. This can be for example an assessment of durability of engine parts. If a problem is defined in such a way that the engine must be examined because the piston rings are damaged, then the task becomes a diagnostic one, i.e. an attempt to identify the cause of defects. If however the problem is presented in a different manner – is there a relation between the damaged piston rings and the type of a lubricating oil – then the experiment becomes a binding one (there will be both qualitative and quantitative dependencies). In case we provide the engine with the rings of another construction, or ones made of another material, and the tests will show improved durability then the experiment becomes a decisive one. Another example is a problem of an improper (incomplete) combustion of the fuel-air mix. In this case, the aim of the research can be the identification of the current state, which makes the experiment a diagnostic one. However, given the same problem, the aim can be formulated in a different manner e.g. what is the relation between the chemical composition of the exhaust gas and the type of injector – the experiment then becomes a binding one. Another example – when you introduce an additive to the fuel and achieve improved combustion quality – the experiment becomes a decisive one. Based on the above-mentioned examples it is evident that it is possible to formulate different aims with the same research problem, and this in turn proves that the experiment can be classified in various ways. As a result, there will be completely different processes of the remaining stages of the research procedure.

The next step in the research programme is the determination of an object. The object in question can be an engine as a whole machine (this mostly is the case) or a system, an assembly or a single element. The research object can also be a process such as the combustion process mentioned above.

The research object can form a simple or a complex structure, e.g. examination of the wear of the cylinder liner, which cannot be separated from the examination of the wear of the piston rings forming a rubbing pair with the cylinder liner, or exclude from the consideration the state of the engine oil, which itself could be a separate research object. Naturally, the degree of complexity of the research object is closely correlated with the aim of the experiment.

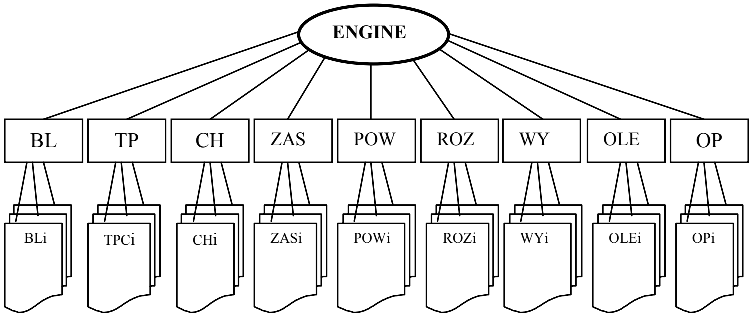


Fig. 1.1 Open systemic structure of a combustion engine
BL – block, TPC – piston-ring-sleeve, CH - cooling, ZAS - supply, POW - air, ROZ – valve gear, WY – exhaust, OLE - lubrication, OP – instrumentation with control unit

Along with determination of the research object, one has to define what physical value relating to the object will be measured. For instance, in order to determine the degree to which the shell of the main bearing is worn out, its thickness can be specified as the measured value, but it can also be its weight or an indirect value in the form of only the thickness of the overlay. You can also use the quantity of the condensate of irradiated particles of a fragment of the ring transferred due to wearing into the lubricated oil and determined using an isotopic method. Attention should be paid to the fact that already at this moment of the procedure, the selection of a measuring methodology is made.

A measuring accuracy should be determined for the measured value so that reliable results and proper conclusions can be ensured. This relates to another assessment which should be carried out, i.e. the specifying of the characteristic of the examined value, i.e. whether it is a constant or a variable value – if it is a variable one then whether the changes occur slowly or quickly, etc. For example, slow changes in the temperature of the cooling agent versus dynamic changes in the pressure occurring in the combustion chamber. At this stage, there may arise a natural barrier of a lack of knowledge, which can only be overcome when adopting proper assumptions. In such cases theoretical considerations may be helpful, provided that the models adopted for calculations are characterised by mathematical significance. Quite often to define the scope of research and to forecast the measured values the experimenter makes use of other researchers' results.

Having formulated in detail the aim of research and having defined the object it is necessary to specify the research assumptions, i.e. to establish the conditions for conducting the research and its limitations and interferences. This stage of the procedure is very difficult, and its completion depends on the experimenter's knowledge and experience.

In the case of research on combustion engines limitations may include, for example, the availability of the investigated object (visualisation of the combustion process or measurement of the temperature of the movable parts of the assembly of a crankshaft, pistons and connecting rods, etc.) Interferences comprise any undesirable and usually uncontrollable influences on the investigated object exerted by its environment and other elements or phenomena. The measurement of volumetric efficiency interfered by atmospheric changes or measurement of wear of the cylinder liner by isotope method interfered by loss of oil resulting from leaky connections may serve as examples of the above.

The elements of the research programme discussed so far, i.e. problem, aim, object, investigated values, measurement accuracy, limitations and interferences, decide about the research methodology, i.e. about selection of the measurement technique and measuring instruments or systems.

First it should be contemplated whether the test is typical, and hence, defined by the standards. If this is the case, the experimenter is required only to ‘follow’ the described methodology. The standard methodology for determination of the main operating parameters of the engine, the so-called performance, i.e. power, torque, effective pressure, etc., can be taken as an example. It is necessary to comply with all the procedures laid down in the standard to ensure repeatability of execution of specific research sequences and proper comparison of the results of the tests conducted for different objects, in different centres, etc. In the case of untypical tests such as investigating the flame front propagation in the combustion chamber, the experimenter has to select the research methodology on his own, taking account of such arguments as the availability of instruments and his financial resources. As soon as untypical tests are transformed into typical ones and satisfactory results are obtained, including their repeatability, the applied methodology can be converted into a standard.

The easiest way for ensuring standardisation of research is to select automatic measurement systems, which start, control, monitor and finish an experiment without any human interference. Such situation becomes increasingly common owing to a fast growth and easy availability of electronic technique, which significantly reduces the share of ‘mechanical’ measurement techniques in research. The reasons for this include the following:

- possibility of conducting research on rapidly changing parameters (e.g. pressure measurement in the combustion chamber),
- significantly lower invasiveness of sensors with regard to the investigated process (thin leaf-type thermocouples used in research on the thermal balance),
- easy data recording, gathering and processing,
- possibility of simultaneous sampling in several or several dozen channels,
- possibility of processing the data in real time (it is extremely important, e.g. for analysing the image making use of laser anemometry while testing the charge movement in the combustion chamber),
- possibility of measurement automation (e.g. exact reproduction of the stage test to measure the toxicity of exhaust gas with simultaneous recording of the results).

The analysis of the possibilities of measurement techniques and their selection for the needs of the research represent the conclusive activities within the first stage, i.e. the experiment programme.

The second stage of the procedure consists in **mathematical experiment design**, which involves the determination of the number of indispensable steps to be made during research, their sequence and optimisation criteria. The definition of the experiment plan requires either the execution of numerous complete conclusive tests or the application of appropriate abstract mathematical tools.

Theoretical bases for optimum experiment design are available [7, 31], however, it is observed in practice that they are not commonly used by experimenters. If any experiment design is done in the research procedure, it is usually of the orthogonal type, which offers the possibility of only qualitative assessment of a given problem.

A stage test can serve as an example of a standard plan of the experiment aimed at assessment of the toxicity of exhaust gases. This test is used to assess the emission into the atmosphere of toxic compounds present in exhaust gases, only on the basis of the tests conducted in the indicated measurement points, assuming the plan hypothesis that if in these points of the engine characteristic the situation is good (qualitative assessment) the same will apply to the whole area of the engine operation.

However, in the research on engines it seems purposeful to popularise the algorithms for optimum experiment planning, taking account of quantitative criteria such as, for instance, a variable frequency of sampling (for example, how far one can go in the concentration of the measurement points in the area of top dead centre and bottom dead centre). Other criteria include the required measurement accuracy, cost of research or tests performed in multidimensional spaces 18, 27, 31] (e.g. drawing up a complete list of variables describing the sources of noise in the engine).

The third stage of the research procedure covers the **execution of an test**, including the design and construction of a test stand, installation of a measurement system, experiment performance and data collection.

A modern laboratory (engine test housing), as a specialist test stand, should meet the relevant requirements already at the design stage. Apart from the construction of the test stand and installation of measurement systems, the activities forming the research execution stage can be, and increasingly are, computer-aided. At this stage we can observe a huge variety of measurement modules, A/C inverter, controllers and data loggers. At present, computer systems fulfil two tasks: measurement registration and control [6, 11, 27, 30].

Such registration can be distinguished by the possibility of synchronisation for the needs of parallel control of selected parameters and the constancy of sampling. Quite often data analysis takes place already at this stage.

Control should first of all ensure the repeatability of the preset measurement sequence, with the highest possible accuracy, and generation of input functions. Control is strictly connected with registration.

The form of experiment execution results from the adopted aim and research assumptions and from the construction of the test stand and measurement system, and it can be a passive, active, computerised active or closed system experiment [27].

In a <u>passive experiment</u> the observer has no possibility of exerting a current impact on the object of research, although it is possible to process data on an ongoing basis, provided that the data processing time is shorter than the sampling time. In experiments involving the engine the assessment of the influence of the fuel injection advance angle on the peak firing pressure can serve as an example of such tests, where there is a possibility of on-going monitoring of indicator diagrams, but any interference with the object can take place only after the measurement-data processing sequence.

<u>An active experiment</u> is characterised by the possibility of exerting an active influence on the object. The measurement procedure ensuring the determination of a load characteristics at the present value of engine rotational speed, adjusted on an ongoing basis, can be taken as an example of such situation.

<u>A computerised active experiment</u> is a research procedure which requires simultaneous execution of two options in real time: data collection and generation of input functions. In this way, for example, the urban test cycle in the fuel consumption testing is carried out, where apart from recording of the values of the measured parameters the input functions concerning the engine load status are generated over a preset time.

<u>A closed system experiment</u>, similarly to a computerised active experiment, carries out simultaneous data recording and generation of input functions, however, it differs from the former one in having each subsequent input function resulting from the reaction of the object in the preceding step of research. This system is used to determine, for example, the wide-open-throttle operating characteristic of the engine, which, after reaching a defined level of the rotational speed value, has a preset input function generation in the form of torque applied to the brake. The stability of these parameters over a defined period of time ‘authorises’ the

control system to change their values, i.e. proceeding to subsequent measurement points of the characteristic.

The next stage of the research procedure comprises the **analysis of the results**, the aim of which is to systematise the collected data and to formulate the conclusions drawn from the research.

The results analysis is carried out on an ongoing basis – during the test – with an increasing frequency, which opens up a possibility of interfering with a research procedure. It results from the possibilities offered by measuring instruments. However, there is a certain risk of the occurrence of information noise coming from the excessive number of measurement data. Such situation can be avoided by a properly planned design and mathematical processing of measurement results.

While planning an experiment a specific number of repetitions of the measurement for the same point in the research area has to be adopted. For each measurement it is necessary to adopt the assumption of physical independence of results, i.e. conducting each measurement according to the best intentions and knowledge and without being influenced by the value of the result of the preceding measurement.

For the results of a series of measurements y_1, y_2, y_3 ... y_n statistical measurement parameters are determined, i.e. mean value, mean square deviation, mean square deviation for a single result and for a series of results and many other parameters, providing the basis for statistical description and drawing conclusions concerning the conducted research [5, 9, 16, 29, 32]. For example, for n – number of results:

- mean value

$$y_{sr} = \frac{1}{n}\sum_{i=1}^{n} y_i$$

- mean square deviation for a single measurement

$$S_y = \sqrt{\frac{1}{n}\sum_{i=1}^{n}(y_1 - y_{sr})^2}$$

- mean square deviation for a series of measurements

$$S_{ny} = \frac{S_y}{\sqrt{n}}$$

- standard deviation for a normal distribution

$$\sigma_y = S_y\sqrt{\frac{n}{n-1}}$$

Measurement results gather around the conventional true value creating the so-called distribution. In the engineering practice there are various types of distributions of results, e.g. normal, exponential, Weibull and t-student distribution, etc., and for these distributions the analysis of the obtained results is carried out [9, 16, 29]. In many cases the experimenter assumes a type of distribution on his own, particularly when the tests are conducted for the first time.

In the assessment of the results a major role is performed by the regression function estimation, i.e. determination of functional dependence on the basis of the input data. Each measurement can be described as in the diagram below

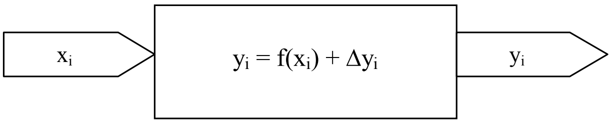


Fig. 1.2 Relation between input and output data
x_i – input value (input function), y_i – output value (measured),
$f(x_i)$ – regression function (unknown), Δy_i – measuring error

The criterion which is most often used for matching the function and the collected results is making the assumption that the expected value of measuring errors approaches zero, i.e. minimisation of $\sum \Delta y_i^2$ function. This method is commonly known as the least squares method or Gauss-Markov model. The calculation procedure employing the least squares method has been described in detail and at present it is a part of the majority of computer programs supporting the engineering work for typical equations with linear characteristics.
In non-linear models the regression function estimation is developed into, for example, the Fourier series, defining the constant and regression range resulting from the experiment design [9, 16].

In the engineering practice we can also find Kernel density estimators of a regression function, based on averaging of the results in smaller ranges and assigning proper significance scores to them, e.g. the Nadaraya-Watson estimator [16].

The so-called stretched thread method or local linear approximation is a very simple method for regression function estimation in a non-linear model. The method consists in linking the important (in the experimenter's opinion) measurement points within the range and putting the function together into a whole on this basis. The Stone and Fan estimator (local linear smoother) is known [16].

The analysis of the results is inseparably linked with the assessment of measuring errors, i.e. the assessment of the difference between the measured and conventional true values. The awareness of the existence of an error makes the experimenter repeat the measurements numerous times, followed by appropriate mathematical processing [31, 32] – Section 1.3.

For a contemporary researcher things are made easier by the application of an electronic system for data processing, with numerous programs supporting the analysis of the results and automatically calculating the final result, presented in the form of a value, equation, table or chart. The final breakdown of the results should include the information about the values of the investigated parameter and the errors accompanying it. Only the results analysis thus conducted can lead to a clear formulation of conclusions drawn from the research. In the conclusions the experimenter should comment on the adopted assumptions, the form of execution and achievement of the aim. The conclusions should offer some suggestions regarding the future work on the investigated problem.

The last stage of the research procedure is the **recording of the results**, which more often than not is combined with the stage of carrying out measurements, during which the data are collected directly on such information carriers as a computer disk, diskette, plotter, printer, etc. This stage, similarly to the analysis of the results, is typical for the contemplated research procedure.

On the basis of the presented material it can be observed that the majority of stages in the research on combustion engines are subject to parameterization and algorithmization. This means that such test can be computer aided, which will lead to far-reaching automation of many activities performed by the experimenter, who, despite all the respect for a human being, is the weakest (fallible) link in the whole research procedure.

Automation of the research procedure will involve, among others, a reduction in the time needed for preparation of experiments (mathematical designing of an experiment) and their completion (computerised data analysis), increased measuring accuracy and in many cases also a qualitative change resulting from technical (electronic) possibilities of making the measurement more detailed.

1.3 Measuring errors and their assessment

It results from the nature of measurements that each of them is burdened by some error attributable to the adopted research methodology, employed measuring instruments, observer's skills, etc. – diagram 1.3.
In general, an error can be defined as a difference between the measured value and the conventional true value, although the latter is an abstract term, established on the basis of the research practice as an expected value.

An error can be classified as an absolute and/or relative error.

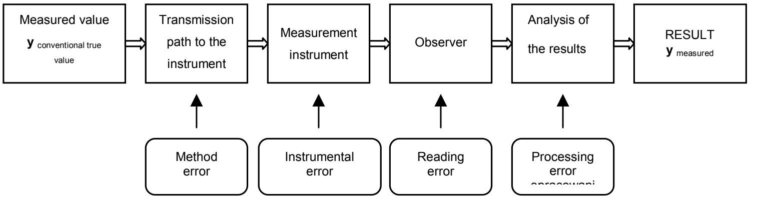


Fig.1.3 Sources of errors [32]

<u>The absolute error</u> is a direct difference between the measured value and the conventional true value of the same tested parameter. The following denotation has been adopted:

$$\Delta y = y_{measured} - y_{conventional\ true\ value}$$

If it is a direct measurement then the maximum absolute error Δy results from the instrumental error Δy_p and the reading accuracy Δy_o and it takes the following value:

$$\Delta y = \Delta y_p + \Delta y_o$$

The absolute error can be positive or negative.

The absolute error assumes the unit of measure of the examined parameter.

<u>The relative error</u> is the quotient of the absolute error and the conventional true value, but due to the difficulties related to a complete identification of the conventional true value, the measured value is often inserted into the formula. The following denotation has been adopted:

$$\delta y = \pm \frac{\Delta y}{y_{measured}}$$

The relative error is expressed in fractions or in percentages – in which case the above value is multiplied by one hundred.
If the absolute error is linked to the of the measuring range (Z), the error is then called a measuring range error, and the formula has the following form:

$$\delta y = \pm \frac{\Delta y \bullet 100}{Z}$$

For an indirect measurement the absolute error of the investigated parameter Δy is a function of directly measured values y_i describing this parameter:

$$\Delta y = \pm \sum_{i=1}^{n} \left| \frac{\partial y}{\partial y_i} \Delta y_i \right|$$

Apart from the classification presented above, there are also other types of classification:

- in respect of the nature of an error
  - systematic,
  - accidental
  - gross
- in respect of the measurement conditions
  - basic,
  - additional
- in respect of the origin
  - method – way of taking a sample for testing,
  - instrument,
  - reading,
  - observer,
  - processing of the results,
  - conditions of testing

Systematic errors, i.e. recurrent errors, are caused by the inaccuracy of the measuring instrument or its calibration, measurement method error or observer, each time making the same error, e.g. a reading error regarding the position of the analogue indicator.

These errors are repetitive, and therefore they can be taken into account as corrections at the determination of real values.

In the case of systematic errors the assessment of measurement accuracy is made on the basis of the knowledge of the maximum absolute or relative errors of the measured values.

The method error is an error in conveying the signal of the measured value from the source to the instrument.

The calibration error can be attributed to the reference instruments (standards) and recording of the instrument characteristic indicating its correct operation. This error is characterised by a constant of a measuring instrument or a function describing the variability of indications.

In the group of systematic errors there can be found the so-called additional errors, resulting from the changes in the reference conditions in comparison to those in which calibration was performed. Their occurrence entails the introduction of the so-called standardising, i.e. mathematical recalculation of the results, taking account of the reference conditions. For example a correction resulting from different atmospheric conditions.

Additional errors include also the so-called zero errors and sensitivity errors. The first one defines the position of the zero point. This error is a constant value for a given instrument, and therefore it can be easily eliminated by entering a correction having the same value, but with an opposite sign.

The sensitivity error is a component part of the indication error of the instrument proportional to indications. The error originates at the stage of designing or construction, and it can be eliminated by regulation or by computation.

The notion of a sensitivity error is related to the notion of an inaccuracy class of a measuring instrument.

The inaccuracy class, or an 'instrument class' in short, means the admissible value of an error at any point of the measuring range in the reference conditions. The error is expressed as a percentage. For example, the conventional classes of 0.1 or 2.5 represent the admissible relative errors of 0.1% and 2.5%, respectively.

$$\text{class} \geq \delta_{dop} \bullet 100 = \frac{\Delta y_{dop} \bullet 100}{Z}$$

Systematic errors include also dynamic errors, resulting from rapidly changing recording and processing of the measured parameter.

Accidental errors are caused by the changing internal and external measurement conditions. These errors include also reading errors, i.e. the difference between the read value and the true value. This error is usually committed by an observer.

The parallax error is an error resulting from an incorrect direction of the projection of the indicating needle on the so-called reading face. Other accidental errors include, for example,
a friction error or a hysteresis error resulting from the changing tribological conditions and mechanical efficiency of the measuring instrument. In digital measuring instruments there may occur the so-called accidental counting error. It arises when, despite the same sampling frequency, the measurement starts at a different research moment – before or directly after the first impulse. As a result, the digital analysis may take into account a different number of measurements in a given series.

To assess an accidental error the theory of probability is applied, similar as in the case of analysing the dependencies between the input and output value.

Gross errors result from the observer's carelessness or lack of knowledge of the method and the measuring instrument. Results burdened with such errors should be rejected (not taken into account during the analysis).

2. BASIC OPERATING PARAMETERS OF A COMBUSTION ENGINE

The combustion engine is the primary source of driving power of many machines and equipment, especially vehicles. Its operational usefulness is defined by parameters of performance, which are used to build characteristics describing the relationship between these indicators.

Basic parameters of combustion engine performance can be divided into those, which are measured directly and those, which are indirect measures. The first group includes such indicators as: engine rotational speed, dynamometer force and arm expressing the torque, the pressure in the combustion chamber, time or mass of fuel consumed, the amount of air sucked in, physical and chemical parameters of exhaust gases, the temperature of operational media or engine components. Indirect measures are: engine output power, efficiency, filling ratio (volumetric efficiency), excess air number, heat of fuel dose, etc. [4, 5, 11, 28].

These measures and indicators are determined as follows.

1. *torque* (M_o) – measured directly on the test stand using a dynamometer

$$M_o = F \bullet b$$

where: F – force applied to dynamometer arm of length equal to „b";
„b" – value measured directly

2. *useful power (brake power)of engine* (P_e)

$$P_e = \omega \bullet M_o$$

where: ω - angular velocity of crankshaft (frequency of rotation) $\omega = 2\pi$ n,
n – engine rotational speed usually expressed as revolutions per minute

Engine power can be distinguished as (among others):

- maximum brake power (highest value of power generated by engine in a short period of time without exceeding the allowable thermal loads of individual elements),
- rated brake power (also called nominal power – power output guaranteed by engine manufacturer to achieve in long term under normal operating conditions),

3. *actual pressure of combustion* (p) – pressure measured directly on engine, through measuring probe placed in combustion chamber, whose changes in the function of changes in the volume of combustion chamber or changes in rotation angle of crankshaft express so called indicator diagram of pressure
4. *mean indicated pressure* (p_i) – calculated (indirect) value of gas pressure in combustion chamber, which performs the same work as a real variable pressure

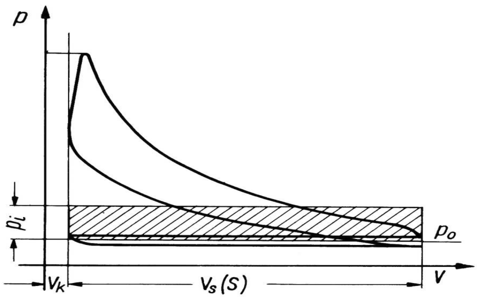


Fig. 2.1 Graphical interpretation of mean indicator pressure [28]

5. *indicated work per cycle* (W_{i1}) – value of work applied to indicated pressure, performed per one engine cycle

$$W_{i1} = p_i \bullet V_s = p_i \bullet A_t \bullet S$$

where: p_i – indicated pressure,
V_s – swept volume of one engine cylinder,
A_t – surface area of the piston crown,
S – piston stroke,

6. *mean effective pressure* (p_e) – ratio of engine performance that defines the ability of engine to perform useful work

$$p_e = \eta_m \bullet p_i$$

or

$$p_e = p_i - p_T$$

where: η_m – mechanical efficiency
p_T – average friction pressure – indicating friction loss due to for example movement of piston rings in cylinder liner or friction of bearings or in valve gear.

7. *useful power of engine* (P_e) – same measure as in point 2, however expressed by a different formula

$$P_e = \frac{V_s \bullet i \bullet p_e \bullet 2n}{\tau}$$

where: V_s – swept volume of one engine cylinder
p_e – brake mean effective pressure
i – number of cylinders
n - engine rotational speed
τ – number of working strokes

8. *fuel consumption per hour (fuel flow)* (G_e) – volume of fuel consumed by engine during one hour. This value can be related to volume of consumed fuel [dm^3/h] or to mass of consumed fuel [kg/h].

$$G_e = \frac{V_p \bullet \rho_p}{t}$$

where: V_p – controlled volume of fuel consumed in time „t”,
ρ_p – fuel density,
t – measured consumption time of a controlled volume of fuel

9. *specific fuel consumption* (g_e) – indirect ratio of engine performance indicating consumed fuel volume (in most cases expressed in grams) in unit of time (in most cases one hour) to generate unit power (kW):

$$ge = \frac{G_e}{P_e}$$

10. *excess air number (λ)* – coefficient measuring the ratio of the amount of air involved in the actual process of combustion (n_{rz}) to the amount of air necessary for full and complete combustion of fuel (n_{teo})

$$\lambda = \frac{\bar{n}_{rz}}{\bar{n}_{teo}}$$

11. *filling ratio (volumetric efficiency)* (η_v) – amount of air sucked into combustion chamber in actual conditions. It can be related to volume (volume of sucked air V_{air} to swept volume of engine V_s) or to air mass (quotient of mass of air supplied to cylinder m_{air} to mass of air filling the entire swept volume of engine m_s).

$$\eta_v = \frac{V_{air}}{V_s} = \frac{m_{air}}{m_s}\eta$$

12. *Theoretical (idea)l efficiency* (η_t) – expresses the ratio between theoretical work (W_t) of thermodynamic cycle and total amount of heat supplied to engine with fuel during that thermodynamic cycle (Q).

$$\eta_t = \frac{W_t}{Q}$$

13. *indicated efficiency* (η_i) – expresses the ratio between indicated work (W_i) and theoretical work (W_t). Indicated efficiency is a measure of loss caused by cooling, friction of charge (charge of combustible fuel and air mixture)

$$\eta_i = \frac{W_i}{W_t}$$

$$\eta_i = \frac{P_i}{\eta_t \bullet G_e \bullet Q_{HV}}$$

where: P_i – indicated power
G_e – fuel consumption per hour,
Q_{HV} – fuel calorific value,
η_t – ideal efficiency

14. *thermal efficiency* (η_c) – expresses the ratio between indicated work (W_i) and total amount of heat supplied to engine during one work cycle (Q).

$$\eta_c = \frac{W_i}{Q}$$

15. *mechanical efficiency* (η_m) – can be expressed, in different ways, as relation between useful parameter and indicated parameter (or ratio of useful output to total input)

$$\eta_m = \frac{L_e}{L_i} = \frac{p_e}{p_i} = \frac{N_e}{N_i}$$

Mechanical efficiency is a measure of any loss on friction and propulsion.

16. *effective (useful) efficiency* (η_e) – ratio of useful work performed (W_e) and total amount of heat supplied to engine during one work cycle (Q).

$$\eta_e = \frac{W_e}{Q}$$

The following formula recapitulates deliberations on efficiency as performance indicator of internal combustion engine:

$$\eta_o = \eta_t \bullet \eta_i \bullet \eta_m = \eta_c \bullet \eta_m$$

Formula applied to tests:

$$\eta_e = \frac{W_e}{G_e \bullet Q_{HV}}$$

Presented below are graphical relations between individual efficiency indicators in the function of relative velocity i.e. ratio of the current velocity and maximum velocity

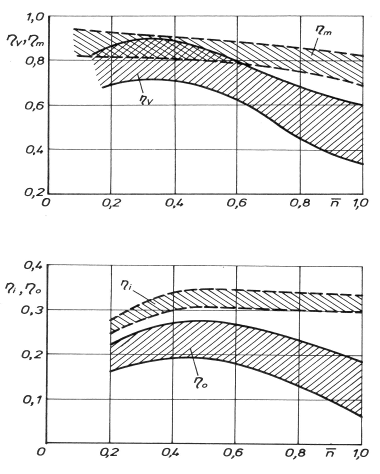


Fig. 2.2. Various efficiencies vs. relative engine rotational speed [28]

17. *torque flexibility ratio* (e_M) – ratio of maximum torque (M_{omax}) to torque of maximum useful power of engine (M_{oN}).

$$e_M = \frac{M_{omax}}{M_{oN}}$$

18. *engine rotational speed flexibility ratio* (e_n) – ratio of engine rotational speed at maximum useful power (n_N) to engine rotational speed at maximum torque (n_M).

$$e_n = \frac{n_N}{n_M}$$

19. *Total flexibility of an engine* (e)

$$e = e_M \bullet e_n$$

Table 2.1 Typical performance parameters of combustion engines of different types [11]

Type of engine	Operating cycle	Compression ratio	Bore m	Stroke to Bore ratio	Engine speed rpm	bmep atm	Power per unit volume kW/dm^3	Weight to power ratio kg/kW	Approx bsfc g/kWh
Spark ignition engines									
Small (e.g. motorcycle	2S, 4S	6-11	0,05-0,085	1,2-0,9	4500-7500	4-10	20-60	5,5-2,5	350
Passenger cars	4S	8-10	0,07-0,1	1,1-0,9	4500-6500	7-10	20-50	4-2	270
Trucks	4S	7-9	0,09-0,13	1,2-0,7	3600-5000	6,5-7	25-30	6,5-2,5	300
Large gas engines	2S, 4S	8-12	0,22-0,45	1,1-1,4	300-900	6,8-12	3-7	23-35	200
Wankel engines	4S	~9	0,57 dm^3 per chamber		6000-8000	9,5-10,5	35-45	1,6-0,9	300
Diesel engines									
Passenger cars	4S	17-23	0,075-0,1	1,2-0,9	4000-5000	5-7,5	18-22	5-2,5	250
Trucks	4S	16-22	0,1-0,15	1,3-0,8	2100-4000	6-9	15-22	7-4	210
Locomotive, industrial, marine	4S, 2S	12-18	0,15-0,4	1,1-1,3	420-1800	7-23	5-20	6-18	190
Large engines and stationary	2S	10-12	0,4-1,0	1,2-3	110-400	9-17	2-8	12-50	180

The relationships between various performance parameters of internal combustion engine are expressed graphically as so called characteristics or maps.

Any theoretical work on development of engine design should be based just on characteristics and in particular, the performance map – see chapter 5.

3. COMBUSTION ENGINE TEST STAND

3.1 Engine test stand

Combustion engine tests at a laboratory stand are called braking tests because of a device called a brake. It is used to force a load proportional to the engine's torque. In this way it is possible to build a characteristic of the engine's parameters and recreate the actual conditions of its operation. To obtain a full picture of the performance, other indications are measured (chapter 2), which entails the use of additional equipment with measuring devices and instruments. These include:

- tachometers,
- devices measuring fuel consumption,
- devices measuring an airflow stream,
- indicators for measuring dynamic pressures in the combustion chamber,
- exhaust gas analyzers,
- meters and auxiliary instruments such as: thermometers and thermocouples, manometers, vacuum meters, barometers, chronographs, aerometers, psychrometers, devices for sounding the engine, etc.),

Fig. 3.1 A test stand
1 – tested engine, 2 – brake with a dynamometer, 3 – drive shaft, 4 – unit to control engine run, 5 – inlet manifold with air flow-meter, 6 – exhaust gas tube, 7 – gauge for measuring fuel consumption, 8 – cooling system, 9 – sensor for measuring engine speed and to mark TDC, 10 – water pipe for feeding test stand, 11 – anti-vibration base, 12 – noise protection wall

Each test stand is also equipped with:

- installation feeding the engine with fuel,
- installation for carrying away exhaust gases,
- installation for cooling the engine, oil and brake,
- wiring system for feeding measuring devices and instruments.

Fig. 3.2 A control room for engine test stand
1 – control unit for engine and brake, 2 – data acquisition system, 3 – fuel consumption gauge, 4 – toxicity analyzers, 5 – window to engine test house

3.2 Brakes for combustion engine tests

Brakes used for determination of a combustion engine's power must be characterized by the following features:

- enable loading the tested engine with torque oppositely directed to the engine's torque,
- ensure carrying away of the generated heat or energy into which the tested engine's operation is converted,
- enable measuring the value of braking torque and the rotational speed of the brake rotor (it is not synonymous with measuring the engine's rotational speed due to the possibility of using a transmission between the engine shaft and the brake),
- ensure the possibility of continuous, sometimes long-lasting, operation of the brake,
- enable obtaining a fast change of braking torque and then a new state equilibrium – the so-called 'ease of control'.

With regard to the way of producing braking torque, dynamometers can be divided into the following types:

- mechanical,
- air,
- hydraulic, commonly known as water brakes,
- electric.

Mechanical and air brakes are the oldest types of devices for loading combustion engines, but they are rarely used today. They can still be met during aircraft or motorcycle engine tests. However, these devices have a few significant disadvantages: low operational stability, strong heating up and hence difficulties in heat abstraction.

Hydraulic (water) brakes

Resistance of a solid body in a fluid medium underlies the operational principle of water brakes. The most important element of a water brake – the rotor – has the shape of a disk with properly shaped blades, which rotates in a water-filled casing. Water particles adhering to the surfaces of the rotating blades are carried away and then thrown outside – to the edge of the disk. In this way the whole mass of the fluid is set in rotary motion and due to being thrown outside, it loses part of its energy as a result of friction against the casing walls. After a short time, a steady flow is produced giving substantial resistance to the rotor's rotation. The strength of hydraulic resistance produces braking torque which is oppositely directed to the rotor's rotation and counterbalances the torque on the brake shaft, to which the combustion engine is connected. Power absorbed in the brake is turned into heat, heating the water flowing through the brake.

The characteristic of a hydraulic brake is shown in the figure 3.3.

Upper line OABC represents braking power for a full load of the brake. Curve OA represents power whose moment developed by the brake has the maximum value. Value M_{omax} is constant and limited by the brake shaft's resistance to torsion. Line BC limits the maximum value of power, conditioned by the range of a temperature increase of the cooling agent. Line CD in turn limits the maximum number of rotor revolutions, conditioned by the limiting load resulting from centrifugal forces. Line OD represents power absorbed when the brake is fully unloaded. The operational area between the abscissa line and line OD of the brake is not controllable, and power is used to overcome frictional resistances: in rotor bearings, on the brake shaft's packing and the so-called residual hydraulic resistances.

Hydraulic brakes are often used at combustion engine test stands due to a relatively simple construction and the possibility of absorbing large energies from the decelerated brake – fig. 3.4 and 3.5

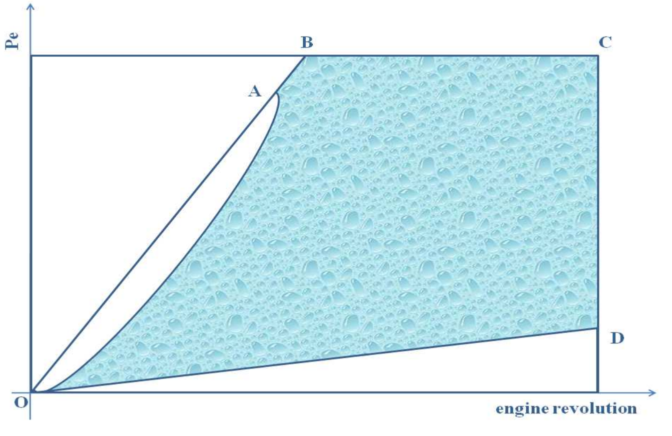


Fig. 3.3 Characteristic of a water brake

With regard to the way of controlling the value of braking torque, hydraulic brakes are divided into:

– water brakes, controlled by the quantity of flowing water, with a water ring flow brakes,

– water brakes fully filled with water under pressure, controlled by means of smooth flaps inserted between the rotor blades and housing blades.

Fig. 3.4 View of the hydraulic brake on the test stand

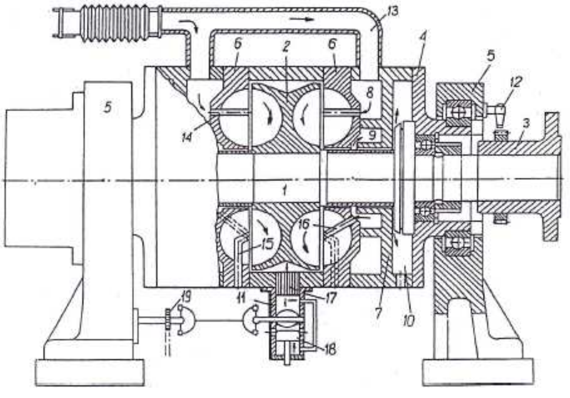


Fig. 3.5 Construction diagram of a hydraulic brake
1- brake shaft, 2- blade rotor, 3- coupling flange, 4- housing, 5- supports, 6- blade disks, 7- partition walls, 8- ring chambers of supplied water, 9- intermediate chambers, 10- chambers carrying away water, 11- control valve, 12- rotational speed sensor, 13- main duct supplying water, 14- duct supplying water to the rotor, 15- venting duct, 16- duct supplying water, 17- control valve inlet, 18- control piston, 19- control valve drive.

Electric brakes

Electric brakes are divided into: electric dynamometers and eddy current brakes.
The advantages of electric dynamometers include:

– possibility of starting the tested engine,
– possibility of partial recovery and use of electric energy,

– easy determination of the tested engine's mechanical efficiency,
– stability of braking torque,
– wide range of rotational speed control.

The disadvantages of electric dynamometers include:
– measurement errors resulting from changes in time of the dynamometer efficiency,
– strong heating up of the dynamometer at higher loads,
– large dimensions in the case of brakes of bigger power rating.

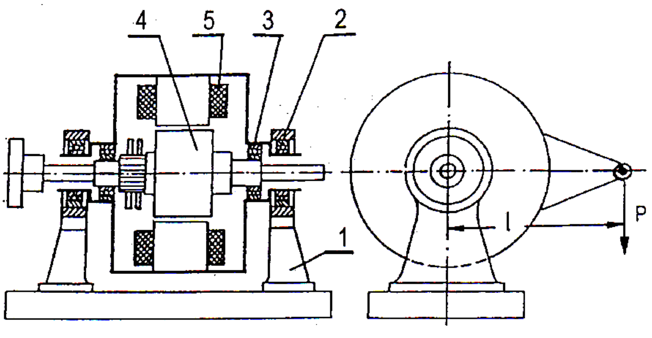


Fig. 3.6 Construction diagram of an electric dynamometer [5]
1 - stand, 2 – stator bearing, 3 – rotor bearing, 4 - rotor, 5 - stator

In eddy current brakes braking torque is produced through eddy currents resulting from changes in the intensity of the magnetic field. A large amount of heat accompanying the operation is carried away by a cooling agent.

The advantages of eddy current brakes include:
– big stability of braking torque and rotational speed,
– low reading error of the measured torque,
– possibility of introducing a cycle of variable engine loads at very short periods of non-stationary states,
– possibility of parallel reading of all results from a control stand,
– possibility of remote controlling and reading of results,
– possibility of using a device protecting the brakes against overloading and damaging.

The disadvantages of eddy current brakes include:
– no direct possibility of recovering energy turned into heat,
– very large cost of investment and service.

The polarity disk (1) with teeth (patterned after a gear wheel) rotates inside a self-aligningly bearinged housing, i.e. the stator (6). In the housing there are also excitation windings (5) and fluid-cooled chambers (7). During the passage of direct current through the excitation winding magnetic field is produced. In the teeth of the polarity disk the magnetic field has a stationary character, i.e. it rotates along with the disk. The rotating polarity disk causes quick magnetization and demagnetization of the disk teeth, as a result of which eddy currents are induced on the cooling chamber walls. The interaction of the main magnetic field and the field coming from eddy currents creates resistance which produces braking torque. Water flowing through the cooling chambers absorbs heat resulting from the exchange of braking energy.

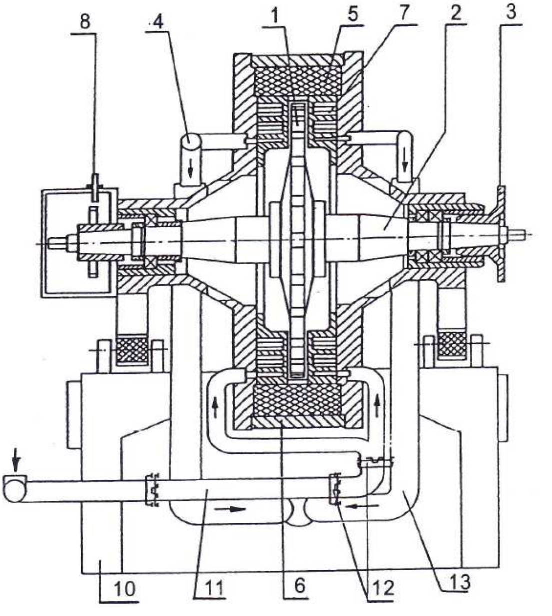


Fig. 3.7 Construction diagram of an eddy current brake [5]
1- polarity disk, 2- brake shaft, 3- coupling flange, 4- water inflow with a thermostat,
5- winding, 6- brake housing, 7- cooling space, 8- revolutions frequency sensor, 10- frame,
11- water inflow, 12- knee, 13- water circulation conduit.

Due to the construction used, lines of the magnetic field run radially in the toothed disk; hence the polarity disk can be thin, and the brake of this construction is characterized by a small moment of inertia.

During the operation of eddy current brakes, one must ensure such flow intensity of water cooling the brake that the temperature of outflowing water is maintained within range 30÷40°C. At higher temperatures, increased furring up takes place in the cooling ducts. Furring up worsens the conditions of heat abstraction and decreases the diameter of cooling ducts. That is why it is so important to service these devices so that the brake is not damaged.

Fig. 3.8 An eddy current brake on the test stand

Figure 3.9 and 3.10 shows diagrams of an eddy current brake's characteristics. The braking torque is controlled by the intensity of the current passing through the winding. This control is very convenient due to the possibility of automating measurements and programming engine loads.

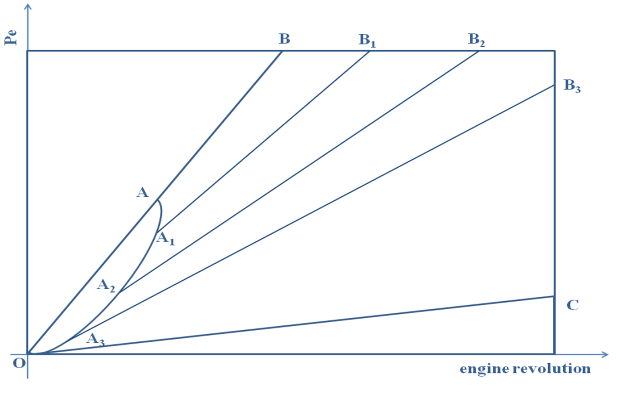


Fig. 3.9 Eddy current brake's characteristics for manual control [5]

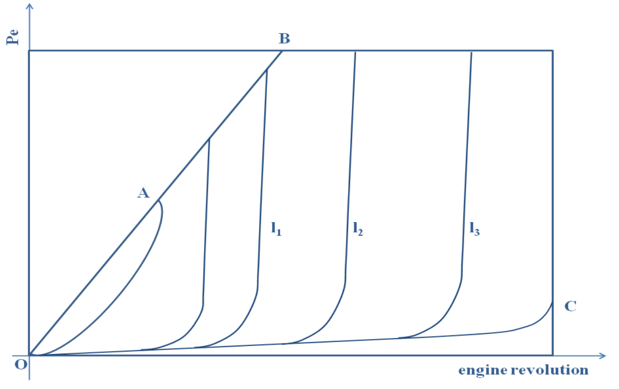


Fig. 3.10 Eddy current brake's characteristics with automatic control [5]

In the above figures, lines AB, A_1B_1, A_3B_3 (fig. 3.9) represent dependence of power on the frequency of revolutions at a constant position during manual control, and lines l_1, l_2, l_3 (fig. 3.10) - during automatic control. Lines OC, the so-called lower limiting curve, and OAB – upper limiting curve, determine the brake's operating area.

Determination of a combustion engine's operating indicators, and above all defining the torque and determining effective power is possible only if the brake has been properly selected. The first parameter to consider is the degree of stability of the dynamometer's operating conditions i.e. the brake's ability to maintain a constant rotational speed at a constant position of the brake and engine settings, and the ability to quickly restore this rotational speed in the event of a momentary disturbance of the torque and the equilibrium of the braking torque.

Fig. 3.11 shows characteristics of engine torque $M_O = f(n)$ and reaction moments of the following brakes: mechanical (M_{mech}), water (M_{hydro}), electric dynamometer (M_{elect}) and eddy current (M_{elw}).

In the mechanical brake, with good lubrication, braking torque hardly depends on the rotational speed. For the water brake, the pattern of braking torque occurs according to a power function with the exponent of about 1.5. The electric dynamometer ensures braking torque in proportion to the rotational speed, and the eddy current brake with electronic control gives a very steep pattern of the braking torque characteristic as a function of relative speed i.e. braking torque as a function of the rotational speed, and to be more precise, as a function of relative speed i.e. a given speed applied to the maximum admissible rotational speed of the brake.

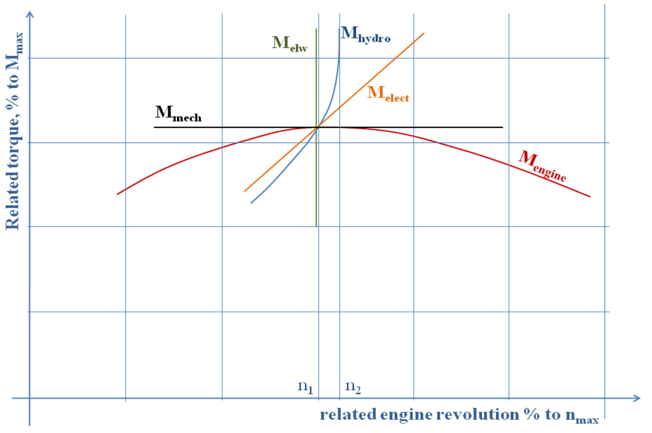


Fig. 3.11 Comparison of the wide open throttle operating characteristic of a combustion engine and torques for different kinds of brakes

Additional braking torque *ΔM*, appearing during a disturbance of equilibrium, leads to a decrease or increase in the rotational speed in relation to the state of equilibrium. From the diagram it follows that the bigger the value of additional torque which occurs during a disturbance of equilibrium, the faster the return to the state of equilibrium, and hence the stability of the brake's operation will be better. It means that the most advantageous brake from the point of view of degree of stability is the eddy current brake. The most disadvantageous is the mechanical brake; what is more, in the event of disturbing its state of equilibrium, additional moments amplifying the disturbance of equilibrium will appear on the side of low speeds. Then, the brake will operate in the so-called state of unstable equilibrium, which will mean the need for its continuous adjustment.

Another important parameter determining the suitability of a brake for a specific type of engine and planned test conditions is the index of "overlapping", determining the degree of overlapping of the engine's and brake's characteristics.

Depending on the predicted plan of tests, the brake's characteristics are compared with the engine's characteristics: wide-open-throttle, full-load or propeller's. If the engine's wide-open-throttle operating characteristic can be inscribed inside the brake's characteristic – curve No. 2 in the figure below, then this brake enables power measurement within the full scope. In the case of the characteristic for engine No. 1 (fig. 3.12) the brake does not make it possible to test this engine, but in the case of engine No. 3 measurements will be incomplete and burdened with large error.

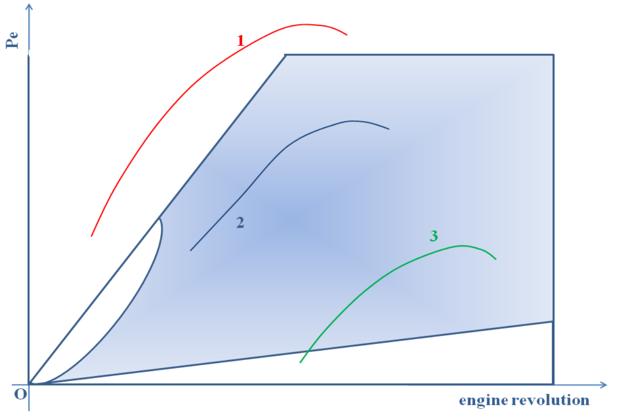


Fig. 3.12 Comparison of an engine's wide-open-throttle operating characteristic with the brake's characteristic
1 – completely incorrect match, 2 – correct match,
3 – incomplete match with a large reading error

Additional recommendations for brake selection:

mechanical brake is used when a small cost of investment, short measurement time, slow rotations and large braking torque are required.

hydraulic (water) brake is used when large braking power, long operation time, good brake stability and high precision are required.

electric dynamometer is used when the use of energy during a long operation time at low power is required.

eddy current brake is used when high measurement precision, automatic control and high operational stability are required.

Choice of the brake determines its calibration. This activity is connected with the need to define random errors such as a friction error or a hysteresis error. Brake calibration consists in applying known torque to an inclinable body of the rotor housing and reading the indications on the dynamometer. The measurements are taken for rising and falling indications. As a result, a diagram is obtained with a distinct hysteresis loop caused mainly by friction – the figure below.

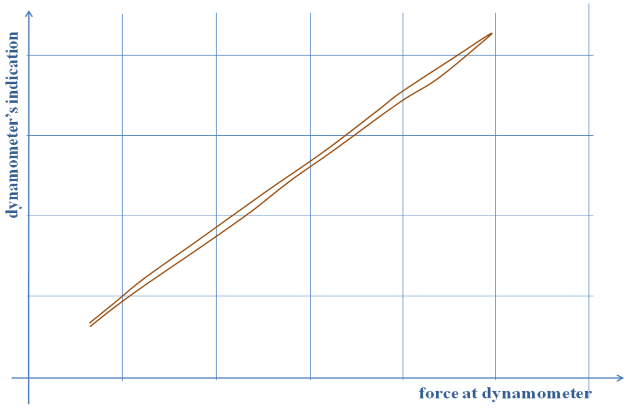


Fig. 3.13 Relation of the actual load of a brake to the dynamometer's indications

Calibration enables determination or verification of the so-called brake constant „K", which is a construction indicator and which must be taken into account in all calculations – e.g. of effective power.

$$K = \frac{R}{F \bullet L}$$

where: K – brake constant,
R – dynamometer's indication,
F = m g – mass of a weight applied at the end of an arm of length „L"

$$P_e = \frac{R \bullet n}{K}$$

where: R – dynamometers' indication,
n – rotational speed,
K - brake constant

If we take into account gravitational acceleration g = 9.81 [m/s^2] and units of intermediate values „m" in [kg], and L in [m] then constant K will equal N/kW min, and the general relation defining the brake constant is as follows:

$$K = \frac{R}{F}\frac{973{,}4}{L}$$

From the above formula it follows that it would be convenient to use a lever of length L = 0.9734 m. To determine brake constant K, averaged values R and F are taken, and then a diagram of relations is created K = f(R).

3.3 Other devices and instruments

3.3.1 Engine's rotational speed measurements

Engine's rotational speed of a combustion engine is one of the most important indicators of the engine's operation and not only as a reference parameter e.g. for speed characteristics, but as a vital diagnostic indicator. In combustion engine tests, typical tachometers are used, such as: mechanical, magnetic, electric, electronic, vibration and stroboscopic.

Among the most popular ones, i.e. electric tachometers, one can distinguish: tachometric, impulse and induction tachometers. Tachometric rev counters consist of two parts: a generator and an indicator which is a voltmeter graduated in rotational speed units.

In impulse tachometers, a spinning element whose rotational speed must be measured, can be used to generate e.g. light or magnetic pulses. In appropriate sensors (e.g. a photocell, a magnetic transducer), these impulses are converted into electric signals which, by means of electronic circuits, indicate the measured values of rotational speeds. In engine practice, other instruments, based on the stroboscope principle, are also used to measure rotational speed. By lighting a rotating element (e.g. the end of a crankshaft) with a mark made on it, you can see a change in the number of flashes with the changes in the apparent rotational speed of the observed mark.

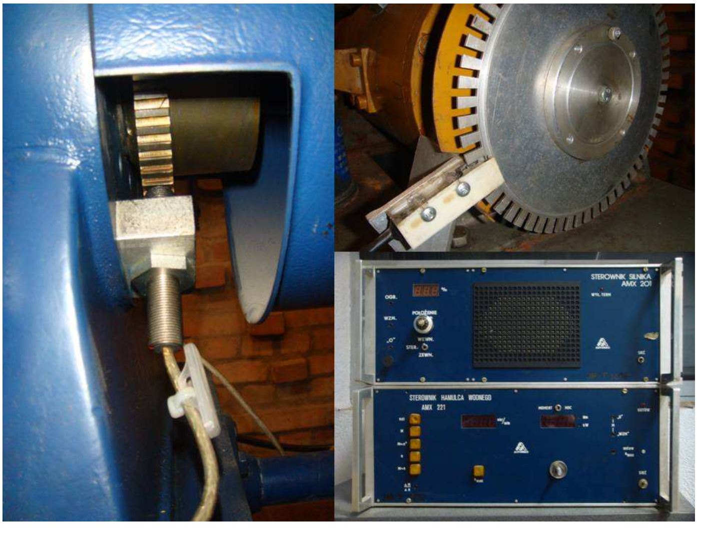


Fig. 3.14 Measuring set for an engine's rotational speed and TDC maker

3.3.2 Fuel consumption measurements

A fuel consumption measurement can be taken by means of the volumetric and/or gravimetric method.

Volumetric fuel consumption is measured by means of a gauge presented below, usually consisting of three interconnected tanks of known volume. The gauge has calibrated marks informing about reaching the assumed values. Determination of fuel consumption consists in measuring the consumption time of a given volume of fuel, and then determining, as an intermediate value, per hour or specific fuel consumption – according to formulas described in chapter 2.

Fig.3.15 View of a fuel consumption gauge at a test stand

The gravimetric method uses the principle of measuring the weight of fuel used during an engines' operation, measured in a device with scales. In this method fuel density is not determined, which is good, taking into consideration that fuel density fluctuates in a wide range, along with changes in temperature. Thus, thanks to the gravimetric method one source of error is avoided.

Subjective errors resulting from time measurement (so-called "an observer's reading errors") diminish with an increase in the measured fuel mass or volume. Then, however, the error resulting from the irregularity of the engine's operation grows.

Contemporary measuring instruments for determining fuel consumption are fully automated, and thanks to suitable software, one can control solenoid valves providing stable filling and emptying of measurement tanks.

Combustion engines with fuel injection systems are equipped with electronic systems of reading the number of pulses of injector openings, and with known values of the volume of the injected dose, they enable ongoing reading of fuel consumption. These methods are based on some activities of statistical character and hence they are not classified as laboratory methods, but only as auxiliary methods.

3.3.3 Temperature measurements

Combustion engine tests as tests of a heat engine must allow for temperature measurements. Therefore, temperature measurements are taken of: the environment, cooling agent, exhaust gases, lubricating oil, cylinder block or head, etc. While taking special measurements, the temperature of the air-fuel mixture or exhaust gases in the combustion chamber is determined, but in this case indirect methods are used, based on theoretical or empirical mathematical relations [17, 35].

The most frequently used measuring instruments are:

– liquid thermometers based on the thermal expansion effect,

– pressure thermometers, in which the relation between liquid pressure, saturated vapour and temperature is used,
– electric thermometers, which include thermocouples, thermistors, resistance thermometers,
– optical pyrometers,
– thermal cameras,
– thermometric paints.

Mercurial thermometers, most commonly found, are widely used for measuring temperatures within the range of - 30°C to + 300°C.

Liquid pressure thermometers, recording the relation between pressure changes and temperature, have been used for measuring cooling agent and oil temperature. Their advantage is the possibility of using long capillaries, which makes it possible to place temperature indicators in a convenient place for reading. The relative error of these thermometers measurements amounts to 2÷5%, so they are not suitable for other, more accurate measurements in engine tests.

Thermocouples have found widespread application in engine tests. They measure the temperatures of:

– exhaust gases,
– head walls, cylinder liners, bearing liners and other engine elements,
– cooling agent in the cooling system,
– lubricating oil in different points of the system.

Metals and alloys used for the production of thermoelements must satisfy the requirements of linearity, stability of the characteristic, high sensitivity and repeatability of measurement results. The most frequently used thermocouples are presented in the table 3.1.

Table 3.1 Thermo-elements most often use [5]

Lp.	Thermo-element plus	Thermo-element minus	Temperature range °C
1.	Platinum - Rhodium Pt – Rh	Platinum Pt	0 - 1300 (1600)*
2.	Nickel - Chromium Ni – Cr	Nickel Ni	0 - 1000 (1200)*
3.	Iron Fe	Constantan Cu - Ni	0 - 600 (900)*
4.	Copper Cu	Constantan Cu - Ni	0 - 300 (400)*
*) temporarily permissible temperature			

Non-contact measuring instruments called pyrometers play an important role in combustion engine tests (like in other machine tests). They enable temperature measurement of a body surface by means of their radiating power which is dependent on temperature. Non-contact thermometers – telethermometers – enable measurement without disturbing the existing temperature field.

Infrared thermographs are based on the same principle. They are equipped with opto-mechanical systems scanning the tested surface, reading not only the temperature, like pyrometers, but also reflecting the shape of the tested body. In this way a thermal image of temperature fields is created, precisely ascribed to specific areas of the tested surface.

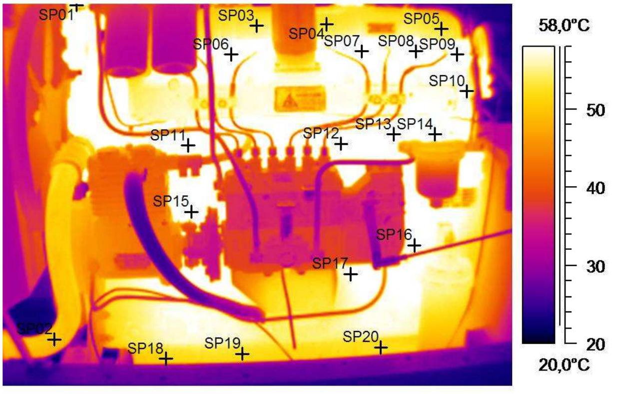


Fig. 3.16 Temperature distribution of an engine block's external surface obtained by means of a thermal camera

In development research of combustion engines, the knowledge of the combustion process, heat exchange in the combustion chamber, etc. plays a vital role. These issues are developed based on complex mathematical methods, but the starting point is always a knowledge of the agent's temperature in the combustion chamber. Due to a closed environment, which is the combustion chamber, and therefore difficulties with placing appropriate sensors, but above all, due to the dynamics of effects occurring in the combustion chamber and lack of appropriate sensors with a very short response time, direct temperature measurement is not possible [11, 15, 34, 35]. This measurement is taken indirectly, and the point of departure is the ideal gas law assuming direct measurement of the temperature and changes in the combustion chamber volume – chapter 2.

3.3.4 Measurements of dynamic pressure inside of combustion chamber

A characteristic feature of piston machines is periodicity of their operation, which causes cyclical changes in the parameters of the gaseous working medium in cylinders, inlet and outlet ducts, crankcases etc.

The thermodynamic state of the medium is determined by three parameters: pressure, temperature and volume. In order to describe the changes which the working medium undergoes in the engine it is enough to know two parameters; the most often are pressure and volume. This is because pressure and volume measurements are easier and more accurate than temperature measurements, which is a derived parameter in thermodynamic calculations. Moreover, pressure distribution in the combustion chamber space is basically uniform, which cannot be said about the temperature. Also, methods and equipment for measurement of dynamic pressures have currently achieved such a technological level that measurements of this parameter have become the basis for obtaining vital information which makes it possible to evaluate the engine's operational process as well as the stress-strain state of the elements of the crankshaft assembly, head etc. as a result of calculations of mechanical and thermal stresses.

Measurements of pressure patterns are taken as a function of time or as a function of geometric quantities such as the angular position of the crankshaft or piston.

The pattern of pressure inside of combustion chamber is called an indicator diagram, for which pressure values are most frequently related to the angular position of the crankshaft. With the correct and complete marking they enable a comprehensive analysis of effects occurring in the tested spaces [1, 3, 21].

Fig.3.17 shows typical diagrams of pressure during combustion and compression stroke for compression-ignition and spark-ignition engines.

The combustion process in compression-ignition engines can be divided into four characteristic periods (Fig. 3.17 a):

- I ignition delay i.e. ignition lag from fuel injection in point 1 to a pressure increase in point 2,
- II propagation of flame from point 2 to 3 with a sudden pressure increase,
- III actual combustion (controlled combustion) from point 3 to 4,
- IV final period i.e. afterburning.

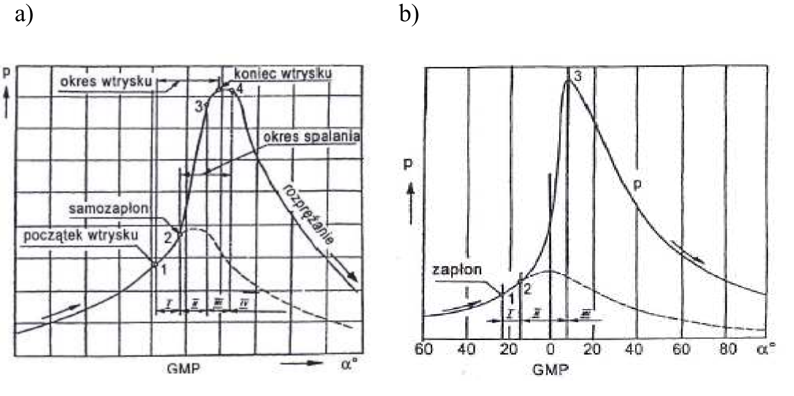


Fig. 3.17 Patterns of pressure changes „p" in the engine cylinder, depending on the crank angle „α" [28]
a) compression-ignition engine, b) spark-ignition engine
[okres wtrysku – injection time, koniec wtrysku – end of injection, okres spalania, - combustion period, samozapłon – self ignition, poczatek wtrysku – start of injection, rozprężania – decompression, zapłon – ignition, GMP – Top Dead Centre]

In spark-ignition engines the characteristic periods are as follows (Fig. 3.17 b):

- I introductory period, called an induction period, starts in point 1 upon the spark-over, and the final period in point 2, when a visible pressure increase occurs,
- II actual combustion, which lasts from point 2 to 3, i.e. from the beginning of a pressure increase to the moment of reaching maximum pressure (propagation of flame in the whole combustion chamber space),
- III afterburning.

Figure 3.18 describes more widely the indicator diagram of a compression-ignition engine. Based on this diagram, the following can be determined:

p_{max}	- peak pressure during combustion,
p_{comp_max}	- peak compression pressure,
α^{o}_{w}	- injection/ignition advance angle,
$\varphi = p_{max}/p_{comp\_max}$	- degree of a pressure increase,

α^o_z - ignition delay angle,
deltap / deltaα - speed of a pressure increase.

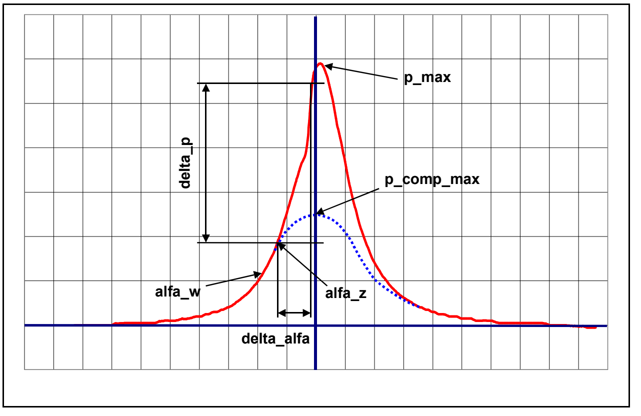


Fig.3.18 Indicator diagram of a compression-ignition engine

The other form, mentioned before, of recording pressure changes is a diagram as a function of time. For an established rotational speed of an engine, the following formula is valid:

$$\alpha^o = 6 \bullet n \bullet \tau$$

where:

n – engine's rotational speed, rpm
τ - time, s

The above relation rests on the assumption of a negligible impact of periodic crankshaft accelerations on pressure patterns in established conditions.

A frequently met form of a diagram of pressure patterns as a function of time is a multiple chart - fig. 3.19, also called a bar chart.

There is a large disproportion between time measured on the reading axis and the duration of a single cycle, hence the lines of pressure increase and decrease overlap. From this kind of charts one can read:

- p_{max} – values of peak pressure of single cycles,
- p'_{max} - value of peak pressure among recorded cycles,
- p'_{min} - value of minimum pressure among recorded cycles,
- $p_{sr\,sp} = \frac{\sum_{i=1}^{i=k} p_{ka\,l}}{k}$ - arithmetic average of pressure (k – number of charts).

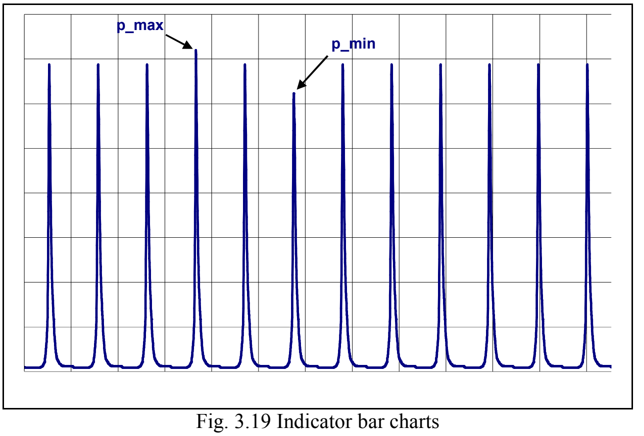


Fig. 3.19 Indicator bar charts

By abandoning the idea of capturing the whole chart, one can record a particularly interesting part of the chart e.g. a diagram of combustion pressure in the TDC area - Fig. 3.20.

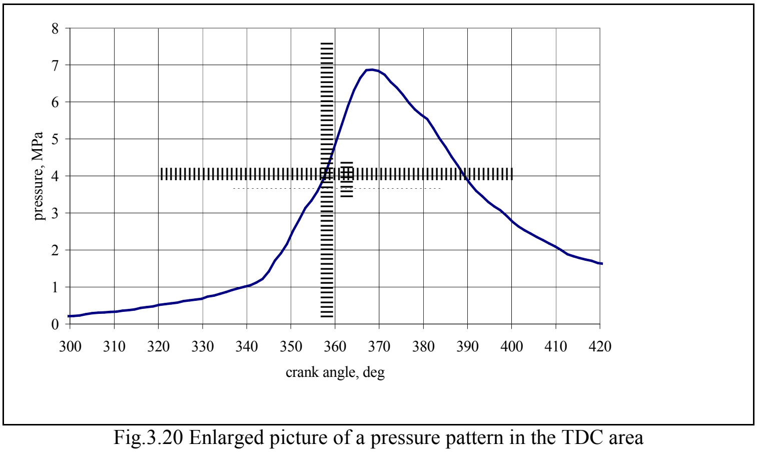


Fig.3.20 Enlarged picture of a pressure pattern in the TDC area

Another form of indicator diagrams are pressure diagrams as a function of piston position (changes in the volume of the combustion chamber) p = f(V), constituting the so-called closed pressure diagrams.

An example of such a diagram is presented in fig. 3.21.

The main purpose of plotting such diagrams is to calculate the average indicated pressure as well as to assess the correctness of the combustion pressure pattern and the compression and expansion pressure patterns. A disadvantage of these diagrams is significant density of the angular scale corresponding to crank angle degrees (CAD) in the area of dead centres (TDC and BDC).

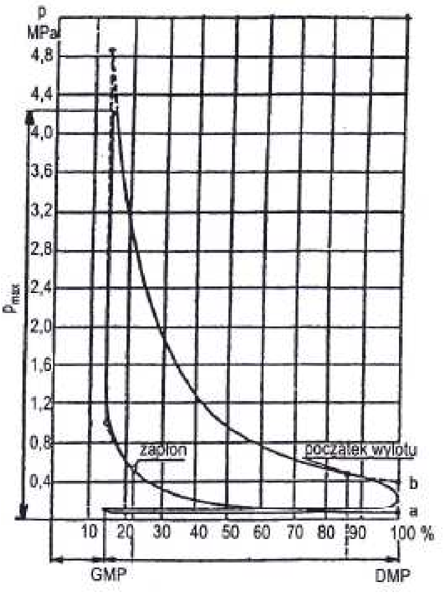


Fig. 3.21 Closed indicator diagram for a spark-ignition engine [28]
[zapłon – ignition, poczatek wylotu – start of exhaust, GMP – Top Dead Centre, DMP – Bottom Dead Centre]

Special diagrams are also made facilitating an analysis of effects occurring in the tested spaces, like in fig. 3.22, where pressure changes during charge exchanges are evaluated.

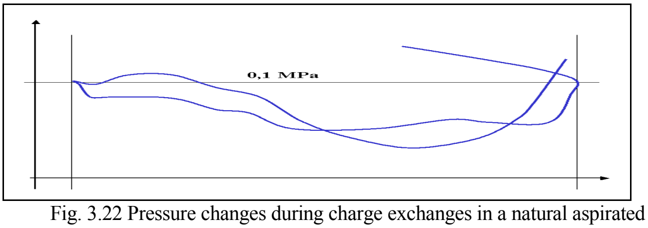


Fig. 3.22 Pressure changes during charge exchanges in a natural aspirated engine

Pressure changes as a function of a specific variable (time, crank angle, piston position) are recorded by means of a special measurement chain. It consists of: indicators, signal transducers, crankshaft position markers, recording systems (measurement data acquisition and processing).

The name indicator comes from the Latin word *indicator*. In engine tests, the following indicators are used:

- mechanical,
- electro-pneumatic (stroboscopic),
- electronic.

Mechanical indicators are suitable only for indicating slow-speed piston machines of rotational speeds below 1000 rpm (e.g. ship engines). They are characterized by big inertia of moving parts and low natural frequency of the small piston - spring system. Besides, they have big so-called clearance volume of the connecting duct and cylinder indicator, which affects the compression degree of the tested engine. Nevertheless, a mechanical indicator can be used for auxiliary testing, such as pressure diagram recording in the cylinder of an engine propelled by another prime mover with a low rotational speed. It makes it possible to evaluate the piston's leaktightness in the cylinder and heat exchange between the working medium and cylinder and head walls. A detailed description of mechanical indicators can be found in professional literature [11, 28].

Electro-pneumatic (stroboscopic) indicators enable measurements in engines of an average rotational speed i.e. up to 2000 rpm, and in special executions – up to 4000 rpm. The best known indicator of this type is the Farnborough indicator. It gives a pressure diagram as a function of crank angle p = f(α); however, only two measurement points correspond to one cycle. Because of this, in order to obtain a full diagram, the measurement should be conducted for a period corresponding to several dozen cycles. This is how an average working cycle is obtained, which, on the one hand, can be seen as an advantage of this indicator, but on the other hand, it is its disadvantage because it is impossible to take measurements and obtain an image of a single cycle in an unsteady motion.

The construction and operation of a stroboscopic indicator is shown in [28].

The use of these indicators is currently very limited because they do not meet the requirements set in programmes of modern and comprehensive tests.

Electronic indicators enable testing of dynamic pressures occurring in different spaces of combustion engines. These indicators cooperate with pressure transducers which generate or modulate specified electric signals, proportional to the working pressure. Fig.3.23 shows an operation chart of an oscilloscope electronic indicator.

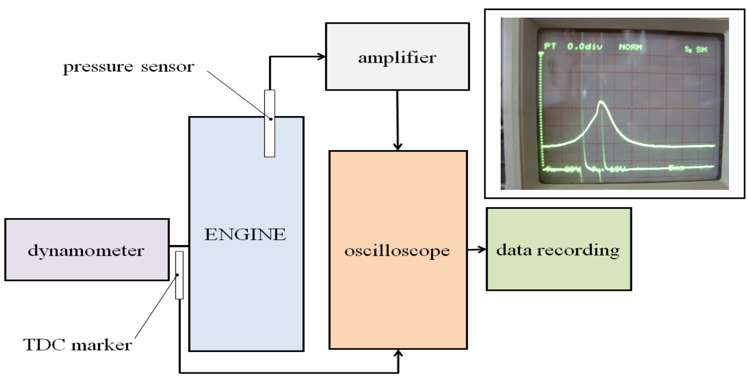


Fig. 3.23 Operation scheme of an oscilloscope indicator

In the tested engine there is a built-in pressure transducer. Pressure pulses are, through a properly selected preamplifier, fed to a Y-Y oscilloscope amplifier, and from there – to deflection plates deflecting the electron beam vertically in the cathode-ray oscilloscope. Horizontal deflection of the electron beam is done either through pulses flowing from the transducer of the piston position to the oscilloscope amplifier – closed-type diagrams p=f(V), or through triangular pulses of a time-base generator built into the oscilloscope p=f(t) – an open diagram. In order to indentify the piston position it is necessary to mark TDC by means of a suitable sensor (TDC timing mark).

During engine tests it is often necessary to obtain an image of two, three or more patterns simultaneously. It is made possible by e.g. a dual-trace oscilloscope. A set of two such oscilloscopes with a common time-base generator makes it possible to track four diagrams at the same time, and by using an electron transducer in the oscilloscope makes it possible to obtain three to five pressure diagrams. A vital advantage of contemporary oscilloscope systems is that they are equipped with a digital processing module with data acquisition. A wide range of functions in which pressure recording systems are equipped facilitates reproduction of an image of low frequency signals as well as provides pattern averaging, eliminating in this way signal interference by using the least squares method or the kernel estimators method.

Pressure transducers

Pressure transducers are basic elements of pressure measurement chains equipped with electronic indicators. Depending on their principle of operation, there are:

- self-generating transducers (active),
- parametric transducers (passive).

In self-generating transducers, under the influence of pressure-related load, an electric signal is generated whose value is measured. The value of pressure is obtained after calibrating the transducer with the whole measurement system by means of a known pressure.

Self-generating transducers comprise piezoelectric transducers, including piezoquartz and electrodynamic transducers.

Parametric transducers powered with alternating or direct current of a known characteristic change it the moment they are loaded. These changes are measured and recorded.

Parametric transducers comprise: resistance carbon, resistance extensometric, inductive, capacitive, and magnetoelastic transducers.

Pressure measurement in the combustion chamber of a piston combustion engine imposes the requirement of measuring the dynamic pressure (a wide range of transmitted frequencies - above 1kHz) of the gaseous medium of a very high temperature (ca. 2000 K). Because of this only few transducers, out of a whole range, have found application. They are transducers: with a cylindrical thin-walled element, with an intermediate element, and transducers with a piezoelectric element.

The most frequently used instruments for pressure measurements in combustion chambers of combustion engines are transducers with piezoelectric elements, particularly quartz sensors.

It results from:

- high measurement stability,
- high resistance to mechanical effects,
- very low inertia which enables a measurement of pressure of frequency 10kHz and more,
- possibility of static scaling.

Unfortunately, their disadvantage is big additional temperature error; that is why in many solutions these transducers come with a cooling system.

The operational principle of a piezoelectric sensor consists in making use of the phenomenon of an electric charge generated on the surface of a crystal plate, compressed in the direction of the so-called electrical axis. The most frequently used piezoelectric element is quartz. Apart from it, similar properties are displayed by crystals of potassium bitartrate, tourmaline or saccharose – used in hydrostatic measurements, and crystals of barium titanate and lead zirconate – for dynamic measurements.

Leading companies researching and producing transducers for pressure measurements in the combustion chamber of piston combustion engines are: AVL, Kistler AG Instrumente, Vibrometer, Hottinger, Honywell.

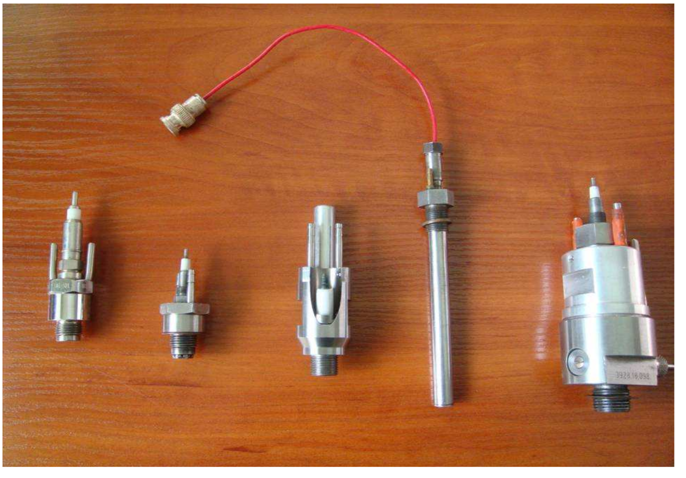


Fig. 3.24 Quartz piezoelectric transducer for pressure measurements

During research it is often necessary to compare pressure changes, which necessitates recording of indicator diagrams. Recording of pressure diagrams in oscilloscope systems takes place by way of photographing the image. To accomplish this aim, full synchronization of the shutter opening time with the course of the light spot is indispensable. It is made possible through a device called a trigger. After being photographed the image is transferred to a millimetre co-ordinate system so that the value can be read.

Modern, computerized measurement systems, which were mentioned at the beginning of the chapter, enable not only ongoing recording of pressure change patterns, but also simultaneous archiving and processing of measurement data.

Computerized recording of values of individual measurement points eliminates the error of handling results, which accompanies manual scanning of oscilloscope photographs and which is the basic advantage of such systems.

The last from among the mentioned elements of a measurement chain is the crankshaft angular position marker, frequently referred to as the TDC timing mark due to the fact that the most important point to define on the horizontal axis of the pressure diagram is the piston top dead centre (TDC) in the tested cylinder. The whole characteristic of the diagram and indirect values determined on the basis of the measured pressure (heat balance, crankshaft assembly load etc.) depend on the correct positioning of this point.

With regard to the operational principle there are magnetic (inductive) and optical (optoelectric) timing marks.

A magnetic sensor is an induction coil which is discharged as it passes through the armature of a special spline, usually situated on the flywheel. Many times there are several such splines and they are located at the same angular distance from each other. It makes it possible to synchronize the excitation system upon passing through the first spline and record the other marks, one of which overlaps with TDC.

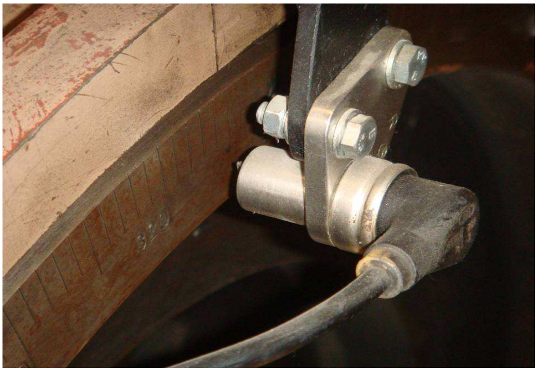


Fig.3.25 Sensor of the crankshafts' angular position

Optoelectric sensors use the principle of breaking a light circuit by a shutter (an obstacle). The above-mentioned special splines on the flywheel can act as the shutter, although in this case their number can be (and most frequently is) many times larger.

Another type of optoelectric timing marks are bar code marks (e.g. CAM Kistler), informing not only about the crankshaft's position, but also about the engine's rotational speed and crankshaft's angular displacement as a result of torsional vibration. In the case of oscilloscope indicators, an impulse from a TDC timing mark can be recorded as one of the streams or timing mark of a break on the pressure "stream".

Like in the whole study, also in this case, ensuring the highest measurement accuracy plays a very important role in determining the tested value. It means the necessity of calibrating the whole combustion pressure measurement chain.

Calibrating consists in determining the relations between the values indicated by the recording system and values set on calibrating instruments.

Pressure measurement systems are subject to static or dynamic calibration. Due to the character of pressure changes in the combustion chamber, an adequate form of calibration is dynamic calibration.

Dynamic calibration enables recording of set pressure and pressure measured by the sensor and maintaining the lines of set pressure jumps. In the case of sensors for which dynamic calibration curves overlap with only a minor error with the static calibration curve (e.g. piezoelectric sensors), static calibration is acceptable. It is substantially simpler in execution.

During such calibration, measurement error depends on the measurement range. Measurement chain calibration is done before and after a measurement, and in the event of long-term tests – also during their duration. The calibration curve enables making corrections of the obtained results, and in oscilloscope systems it takes place while handling results. For computerized measurement systems there is a possibility of encoding the measured corrections to a storage block and allowing for them in real time – during the measurement.

Measurement accuracy of indicated pressure is determined by a number of measurement aspects. These include: calibration of the measurement chain, sensor position in the combustion chamber, filtration of the measurement signal (elimination of disturbances), determination of the position of inner dead centre - TDC, sampling frequency of the indicator diagram.

Badly performed calibration leads to wrong conclusions about the measured pressure. For example, if one wanted to measure simultaneously peak pressure of about 10 MPa and pressure of - 0.1 MPa with 1% accuracy, one would need to have equipment of accuracy class 0.01, which is extremely difficult, and in computerized systems at least 18-bit cards would be necessary. What it means practically is that during measurements of high pressures, values of pressure from the lower area of the indicator diagram must be treated as approximate values and the other values as very accurate [1, 14, 33].

Proper calibration can be spoilt by wrong interpretation of results, which e.g. does not allow for the way of mounting the sensor. Fig. 3.26 shows different ways of mounting a pressure sensor resulting from sensor type and technical possibilities of mounting it in the engine.

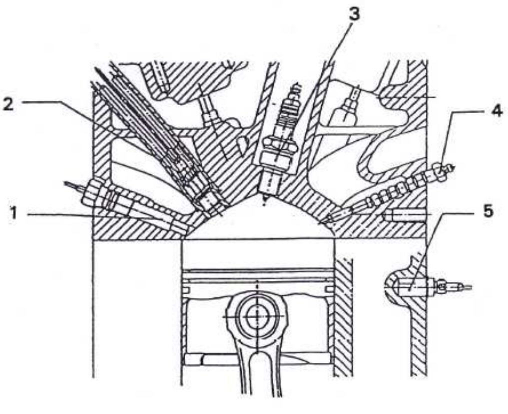


Fig. 3.26 Examples of different ways of mounting pressure sensors in the combustion chamber (Kistler Inc. Service Instruction)

The most advisable way of mounting the sensor is placing it in the head so that the measurement membrane is at the level of the upper edge of the chamber - Fig. 3.27.

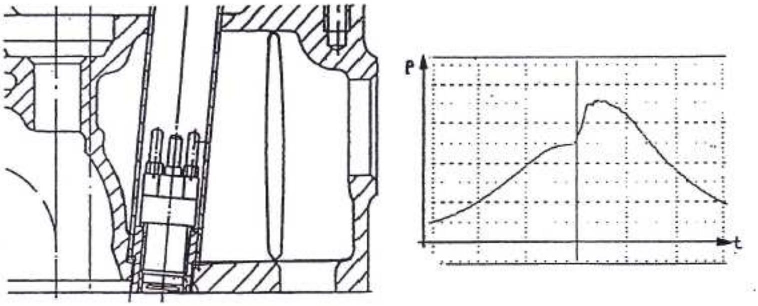


Fig. 3.27 Recommended way of mounting the pressure sensor (Kistler Inc. Service Instruction)

However, it very often happens that due to the limited mounting space or additional difficulties (water or oil ducts etc.) the sensor is mounted differently than in fig. 3.27. The sensor is mounted through additional openings and connecting ducts, which unfortunately contribute to further measurement errors, resulting from gas column oscillations in these ducts - fig. 3.28.

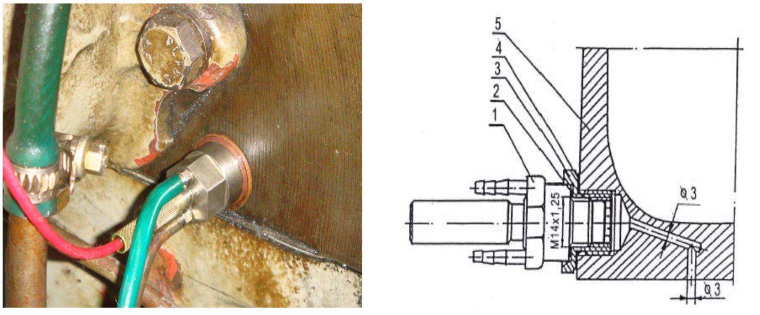


Fig. 3.28 Mounting a pressure sensor with an intermediate duct
1 – transducer's cooling system, 2 - piezoquartz sensor, 3 – reducing nut,
4 – a packing washer, 5 – engine block

Pressure in the cylinder is constantly changing, and the curve on the indicator diagram, reproducing this process, has an undulating pattern. This undulation is explained by: variability of the speed of load flow, variability of the flow cross-section of the valves, difference in the speed of the combustion process front and piston movement. When lacking suitable equipment, the above-mentioned fragments of the diagram are replaced with straight lines and roundings in places of contact of these fragments. This method, called measurement signal filtration, is performed by means of aggregation (concentration) of adjacent signals or value weighing e.g. according to standard deviation of the measured values. Also, only extreme values of the measured range are often determined and then connected with straight lines.

Fig. 3.29 shows pressure diagram: actual and after filtration by means of weighing averages of period equal 5 for engine SW680 at rotational speed n = 2200 rpm and full load - M_{omax}.

Another cause of pressure measurement errors can also be inaccuracy of determining the TDC point on the indicator diagram. It is particularly vital for further handling of results which take into account not only pressure changes, but also changes in the combustion chamber space, e.g. thermal calculations. In this situation overrated or underrated values of the combustion function are obtained and, consequently, diverse values of the working medium temperature (T_1, T_2 - fig. 3.30).

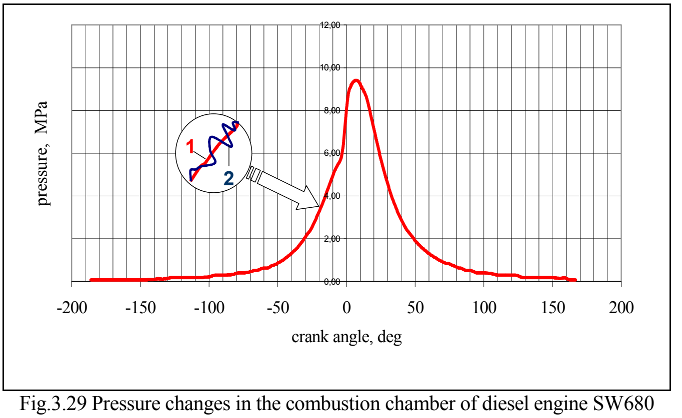


Fig.3.29 Pressure changes in the combustion chamber of diesel engine SW680
at n = 2200 rpm and full load
1 – curves corresponding to the pattern after signal filtration,
2 - curves corresponding to the actual pattern.

Fig. 3.30 shows patterns of thermodynamic parameters, describing thermal load of engine SW400 for a wide-open-throttle operating characteristic n = 1600 rpm. Values with index "1" refer to a pattern with a precisely marked TDC point, and patterns with index "2" refer to TDC "shift" by minus 2 crank angle degrees (CAD). Such a "shift" of the indicator diagram can result e.g. from the difficulty in interpreting the image of the TDC timing mark (triangle, rectangle on the oscilloscope screen or computer monitor, beginning or end of a break, etc.), recorder's parameters, sensor's inertia, etc.

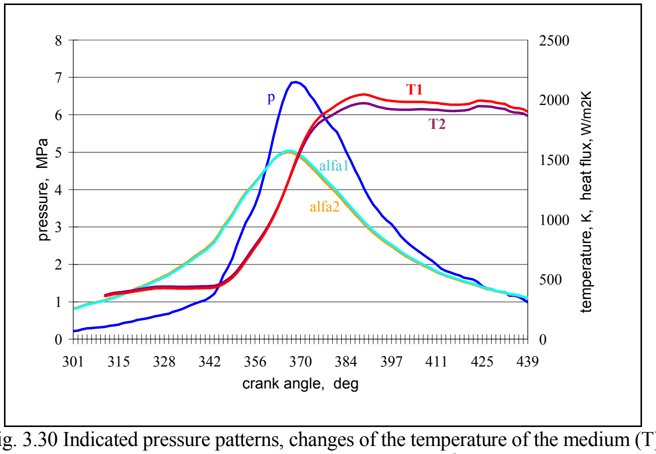


Fig. 3.30 Indicated pressure patterns, changes of the temperature of the medium (T) and heat flux (α) caused by TDC "shift" by minus 1 CAD ^{o}OWK for engine SW 400
1 – precisely determined TDC point, 2 – patterns relating to TDC shift by minus 1 CAD

In this figure, changes of the temperature of the working medium are also marked, which is calculated on the basis of the measured pressure, taking into consideration the piston's position – the TDC timing mark. Incorrect TDC marking in this case made a difference of the average temperature T of the medium by as much as 72 K (!) (4.4% of maximum value).

Out of the considered aspects determining the suitable construction of an indicator diagram, correct sampling frequency seems to be the obvious and least questionable factor because it is known that sampling with the frequency of 10.000 Hz or 100.000 Hz will ensure more reliable results than a measurement of 1000 samples per second. Depending on the engine's rotational speed it gives measurement intervals from 0.5 CAD or 5 CAD, and even 10 CAD. Of course the latter values, especially in the TDC area, are not acceptable if only due to the possibility of omitting the TDC timing mark. Too high frequency of signal sampling carries the risk of appearance of the so-called pressure peaks resulting from pulse disturbances or momentary overflows of the storage block; hence, the previously-mentioned signal filtration. Both these activities, i.e. signal filtration and selection of sampling frequency must be done very carefully and delicately. In order to avoid gross errors, measurements must be taken many times, which should not pose any problems with the computer-collected and processed data.

An analysis of accuracy of pressure representation in the full operational cycle of a combustion engine indicates the significance of the measurement chain calibration, way of mounting and accurate representation of the TDC position. For other cases, i.e. signal filtration and sampling frequency, their impact is smaller and statistically insignificant for the considered cases. However, a hypothetical superposition of errors resulting from: inappropriate signal filtration, incorrect determination of the TDC position and too low sampling frequency can lead to differences between the determined parameters of even up to 10 %.

3.3.5 Gas flow rate measurements

The measurement of gas flow rate is a parameter necessary to determine relevant ratios of operation of the internal combustion engine, i.e. the charging efficiency and the excess air ratio. The measurement is performed by means of flowmeters, which include: anemometers, thermoanemometers, rotameters, constriction and float flowmeters, and impact tubes, such as Prandtl tube.

In practice, tests of internal combustion engine typically use vane anemometer and constriction flowmeters. Although thermoanemometers constitute commercial solutions of many engine manufacturers, they primarily ensure qualitative measurements. Laboratory tests require high accuracy, which is not guaranteed by this type of sensor.

<u>Vane anemometer</u>

A vane anemometer is easy to use. A change of the rotational speed of the rotor into electric signal, and then into the value of the flow rate, enables observation of momentary flow rate values. A vane anemometer is also characterised by high versatility in the case of engines with variable displacement.

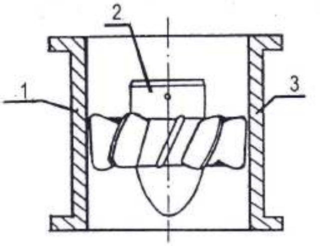


Fig. 3.31 Design diagram of an anemometer turbine
1 – casing, 2 – rotor, 3 - vanes

Constriction flowmeter (in short: orifice) ensures measurement of the flow rate of gas or liquid using the principle of proportionality of mass or volumetric flow rate to differential pressure caused by a calibrated orifice. The orifice is built into a measuring pipeline of strictly defined dimensions.

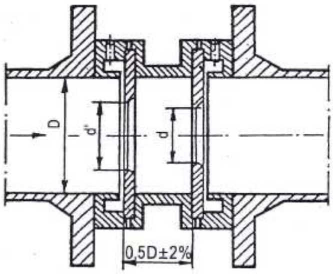


Fig. 3.32 Design diagram of a measuring orifice plate (constriction flowmeter)

In an internal combustion engine, due to the nature of its work (power strokes), flows occur in pulses, which may affect the accuracy of measurement. Therefore, the systems with orifice are supplemented with a large-volume vessel to ensure levelling of those pulses. Hence the name of that vessel – compensatory. On the other hand, small air flow rate on the engine inlet forces the use of a double-orifice system, which ensures measurement of the flow rate for Reynolds numbers smaller than the limit value.

Mass flow rate (intensity) is expressed with the following formula

$$m_P = \alpha \bullet \varepsilon \frac{\pi \bullet d_1^2}{4} \sqrt{2 \bullet \Delta p \bullet \rho_1}$$

whereas volumetric flow rate with:

$$V_{air} = \alpha \bullet \varepsilon \frac{\pi \bullet d_1^2}{4} \sqrt{2 \bullet \Delta p \bullet \nu_1}$$

where:

α – flow number,
ε – expansion number,
d_1 – orifice diameter
Δp – differential pressure
ν_1 – specific volume of air in normal conditions

number of expansions is determined according to the following formula

$$\varepsilon = 1 - \ (0{,}3707 + 0{,}3184 \frac{d_1^4}{D^4}) \bullet [1 - (1 - \frac{\Delta p}{p_1})^{k-1}]^{0{,}935}$$

where:

D – diameter of the inlet channel
p_1 – absolute air pressure at the orifice
k – isentropic exponent of the tested gas

Under set thermodynamic conditions and assuming invariability of the number of flows and expansions, we can reduce the formula for the volumetric flow rate as follows

$$V_{air} = C\sqrt{\Delta p \bullet v_1}$$

where: C- orifice constant

When we have the flow data, we can indirectly determine the charging efficiency – chapter 2.

$$\eta_v = \frac{V_{air}}{V_s} = \frac{m_{air}}{m_s}$$

The volumetric efficiency of air fill for the whole engine, working at a specific rotational speed for one hour, should be expressed as follows:

- for a two-stroke engine

$$\eta_v = \frac{V_{air}}{60 \bullet n \bullet i \bullet V_s}$$

- for a four-stroke engine

$$\eta_v = \frac{V_{air}}{30 \bullet n \bullet i \bullet V_s}$$

where:

V_{air} – volumetric air flow rate reduced to normal conditions, m3/h
V_s – displacement (swept volume) of a single cylinder, m^3,
n – engine's rotational speed, rpm
i – number of cylinders

In the engineering practice engine tests can include determination of the relation of the charging efficiency to the rotational speed or the load through the whole operating range.

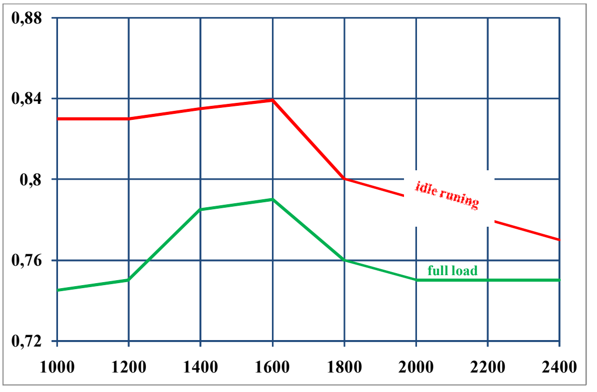


Fig. 3.33 Graph of the dependence of volumetric efficiency (y-axis) on rotational speed (x-axis) for a natural aspirated diesel engine
1 – idle running, 2 – full load

Ability to determine the air flow rate and the knowledge of the volumetric efficiency enables determination of another important work index, i.e. excess air ratio λ – chapter 2.

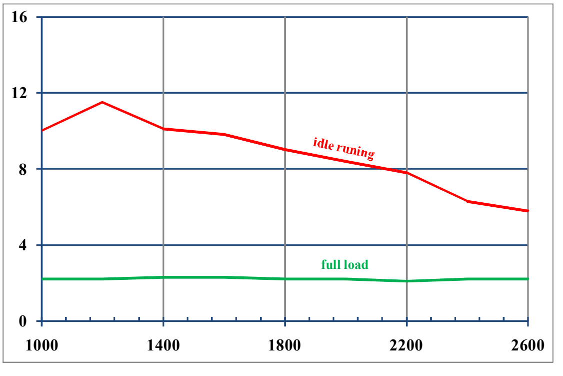


Fig.3.34 Change in the excess air ratio (y-axis) together with the engine's rotational speed (x-axis) for a supercharged diesel engine
1 – idle running, 2 – full load

This ratio characterises the efficiency of the combustion process based on the ratio of the amount of air taking part in the actual combustion process to the theoretically needed amount to ensure complete and perfect combustion.

Excess air ratio is determined taking into account the following formula based on the ideal gas law (the Clausius-Clapeyron equation)

$$\lambda = \frac{\eta_v}{B_o} \bullet \frac{0{,}21}{n_{min}} \bullet \frac{V_s}{MR} \bullet \frac{p_{ot}}{T_{ot}}$$

$$B_o = \frac{0{,}12 \bullet \rho_{fuel} v_x}{n \bullet \tau_x \bullet i}$$

where:

η_v – volumetric efficiency of a single cylinder
B_o – fuel charge per cycle, kg
n_{min} – the minimum amount of oxygen needed for complete and perfect combustion of 1 kg fuel, kmol.kg
V_s – displacement of a single cylinder, m^3,
MR – universal gas constant, J/)kmol*K)
p_{ot} – ambient pressure, Pa
T_{ot} – ambient temperature, K
ρ_{fuel} – fuel density, g/cm^3
v_x – fuel dose volume, cm3
t_x – fuel dose consumption time v_x, s
n – engine's rotational speed, rpm
i – number of cylinders

The value of excess air ratio is determined indirectly, therefore, to ensure high accuracy, it is very important to minimise the values of systematic and random errors. Systematic errors can be reduced e.g. by making sure a high-quality orifice is used to measure the intensity of air flow; on the other hand, random errors can be reduced by accurate readout of the consumption time of a fuel charge as well as measurement of the volume of that charge.

3.3.6 Measurements of toxicity of exhaust gases

For dozens of years, development of internal combustion engines has been subordinated to the broadly defined ecology of the environment. Therefore, we should include information about tests of toxic components of exhaust gases. Tests of exhaust gases have two primary purposes – assessment of risk for the human body and determination of the level of toxicity that must be observed by the producers and during use. Unfortunately, exhaust gases, despite theoretical considerations of their non-toxicity (CO_2 and H_2O), are highly toxic and their share in the environmental pollution is estimated to be at a level of between 20% and 40% (depending on the region of the world) of the total pollution [23, 24]. Fuel combustion in a real engine is incomplete and imperfect, which means that, in addition to the above-mentioned CO_2 and H_2O, other products appear, such as: CO, C_nH_m, and C_nH_mO.

Although carbon dioxide is not directly harmful to the human body, it is now widely identified as a chemical compound responsible for global warming, and, therefore, it is also undesirable. Water, on the other hand, which is present in exhaust gases in the form of vapour, interacts with trace amounts of sulphur in the fuel, so it has corrosive effect and, on a climatic scale, is partially responsible for acid rains.

Nitrogen, sucked with the air, ceases to be an inert gas in the engine and, at very high temperatures in the combustion chamber, changes into harmful nitrogen oxides (NO_x).

Combustion of diesel fuel in compression-ignition engines is accompanied by soot formation, which, although considered non-toxic, becomes toxic due to high surface adsorption as toxic particulate matter.

The following table shows the average quantitative composition of exhaust gases of car engines.

Table 3.2 Composition of exhaust gases
in spark-ignition (SI) and compression-ignition (CI) engines

Exhaust gas components	Engine type			
	SI		CI	
nitrogen	74 - 77	% vol.	76 – 78	% vol.
oxygen	0.3 - 8.0	% vol.	2 – 18	% vol.
water vapour	3.0 - 5.5	% vol.	0.5 - 4.0	% vol.
carbon dioxide	5 - 12	% vol.	1 – 10	% vol.
carbon monoxide	5 - 10	% vol.	0.01 - 0.5	% vol.
nitrogen oxides	0 - 0.8	% vol.	0.002 - 0.5	% vol.
hydrocarbons	0.2 - 3.0	% vol.	0.009 - 3.0	% vol.
aldehydes	0 - 0.2	% vol.	0.001 - 0.009	% vol.
sulphur dioxide	0.01	% vol.	0.04	% vol.
soot	0 - 0.1	g/m^3	0.01 - 1.1	g/m^3
lead compounds	60	mg/m^3	none	

The quantities of toxic exhaust components are defined by diving the volume or weight of the examined component by the volume of the mixture in which its participation is considered. The volume ratio is called volume share or concentration, and the ratio of weight to volume is called weight concentration.

Volume concentration is expressed as:

– volume percentage (% vol.),
– parts per million (1% vol. = 10^4 ppm),
– parts per billion (1% vol. = 10^7 ppb).

Weight concentration is expressed in mg/dm^3 or mg/m^3. Tests conducted solely to determine harmfulness to the human body additionally use weight concentration related to 10m^3 of the air, i.e. the maximum amount of the air breathed in by a man during an eight-hour working day. On the other hand, the intensity of presence of a particular exhaust component is determined by its emission. It is stated in g/h or as a specific emission expressed in mg/kWh or in g/km for traditional applications.

Qualitative assessment of the process of combustion in the engine is obtained by expressing the quantity of toxic exhaust component in relation to 1 kg of used fuel, adopting the unit of g/kg [5].

Comparability of the test results was ensured through research studies that created operating conditions of the engine corresponding approximately to the average operating conditions. Those conditions had been previously described statistically, therefore we can come across e.g. long-term research tests or urban cycles. Furthermore, the measurement conditions of exhaust toxicity are defined (by developing the research methodology) and the limit values of toxic components per cycle are determined.

Depending on the measured size, tests (analysis) of exhaust gases use various devices, whose operating conditions are based on diverse phenomena ensuring measurement accuracy.

These are:

- absorption of a part of the infrared spectrum (NonDispersive InfraRed analysis – NDIR) or the ultraviolet spectrum to determine the concentrations of: CO, CO_2, CH_4, NO, SO_2, and O_2;

- Flame Ionization Detector (FID) analysis by the introduction of hydrocarbon particles to the hydrogen-air flame to determine total hydrocarbons (HC); the mixture is separated into individual components with the use of chromatography gas (GC);

- emission of electromagnetic radiation accompanying the reaction of nitrogen oxide NO with ozone O_3, called chemiluminescence (Chemiluminescent Detector - CLD), to determine the concentrations of NO and NO_2;

- paramagnetic properties of an oxygen particle to determine O_2 concentration – MPD analyser;

- properties of electrochemical cells to determine concentrations of: CO, CO_2, NO, SO_2, and O_2.

<u>NDIR analyser</u>

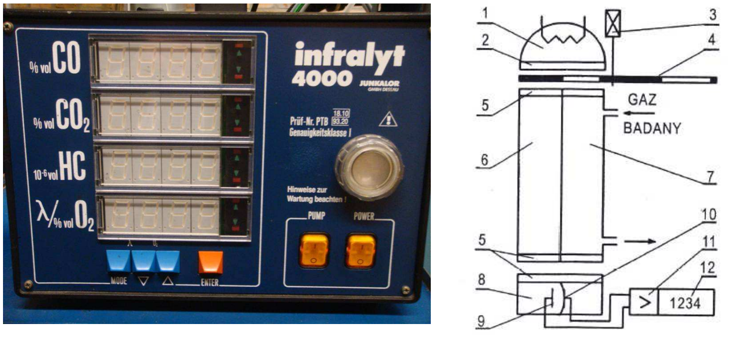


Fig.3.35 View and operation diagram of a nondispersive infrared (NDIR) analyser [5]
1 – infrared heater, 2 – optical filter, 3 - engine, 4 - diaphragm, 5 – quartz window, 6 – comparative cuvette, 7 – measurement cuvette, 8 - detector, 9 – stationary plate of a capacitor, 10 – membrane with movable capacitor cover, 11 – signal amplifier, 12 - meter
Gaz badany – tested sample of exhaust gas

Flame ionizing detector (FID) analysers

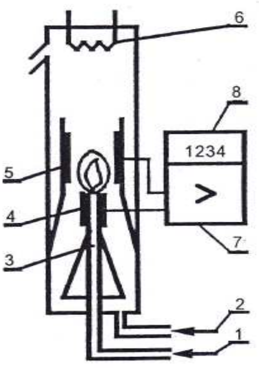


Fig. 3.36 Operation diagram of a flame ionizing analyser [5]
1 – supply of a hydrogen-nitrogen mixture and the tested gas, 2 – air supply,
3 – hydrogen-oxygen burner, 4 – cold electrode, 5 – hot electrode, 6 – igniter's spiral,
7 – signal converter, 8 - meter

Gas Chromatograph (GC)

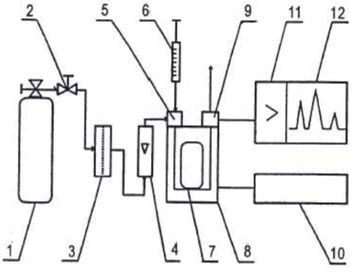


Fig. 3.37 View and operation diagram of a gas chromatographer [5]
1 – carrier gas cylinder, 2 - control, 3 - filter, 4 - flowmeter, 5 - dispenser,
6 – syringe for gas sample administration, 7 - column, 8 – thermostatic chamber,
9 - detector, 10 – temperature control, 11 - amplifier, 12 - recorder

CLD analyser

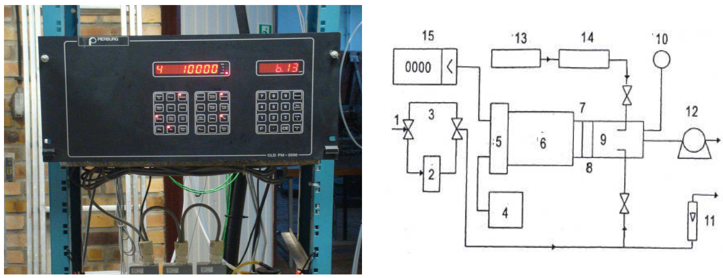


Fig. 3.38 View and operating diagram of a chemiluminescence analyser [5]
1 – supply of the analysed gas, 2 – nitrogen dioxide converter, 3 – shunt cable,
4 – high voltage supply, 5 - electrometer, 6 – photomultiplier tube, 7 – optical filter, 8 – quartz window,
9 - reactor, 10 – vacuum gauge, 11 – flowmeter, 12 – vacuum pump, 13 – ozone generator, 14 – source of oxygen.

MPD analyser

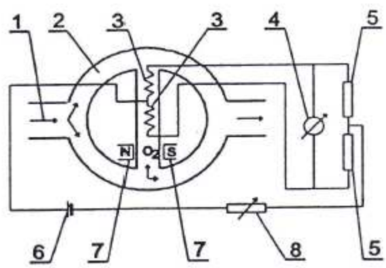


Fig. 3.39 Operation scheme of paramagnetic analyser for determining oxygen concentration [5] 1 – tested gas inlet, 2 – two-branch circular channel, 3 – electric bridge resistors in the transverse channel, 4 - meter,
5 – external resistors of the electric bridge, 6 – power cell,
7 – permanent magnets, 8 – control resistor

A separate group of tests of compression-ignition engines consists of **measurements of exhaust smoke**, commonly known as **soot studies**. The transparency-reducing smoke in exhaust gases is caused by soot, liquid organic compounds, and particles of unburned fuel and lubricating oil present in the gases.

Depending on the colour of exhaust gases of compression-ignition engines, we can distinguish the following types of smoke:

– white smoke – occurs at the start and small load of the engine and is a results of condensation of water vapour and the presence of increased amount of unburned hydrocarbons;
– blue smoke – given to exhaust gases by particulate matter, a product of breakdown, partial oxidation, and coking of hydrocarbons, which constitute components of the lubricating oil that penetrates to the combustion space;
– black smoke – caused by the presence of soot – noticeable when the soot concentration stands at 100÷120 mg/Nm^3 and almost black when the concentration

stands at 600 mg/Nm^3.

Black smoke informs about incorrect combustion process caused by inappropriate control settings (fuel charge, injection advance angle, injector opening pressure) or the condition of the engine and the injection equipment (partially blocked air filter, low fuel pumping pressure of the injection pump, leaky injector sprayer).

Smoke density is a value that uniquely characterises exhaust smoke. It defines the weight of soot present in unit volume of exhaust gases under normal conditions – hence, the mg/Nm^3 measure. In the engineering practice there is still no instrument that directly measures smoke density, therefore smoke is assessed by indirect methods, determining:

- soot content in exhaust gases (filtration smokemeters),
- smoke on the basis of exhaust opacity (light obscuration smokemeters).

Filtration smokemeter

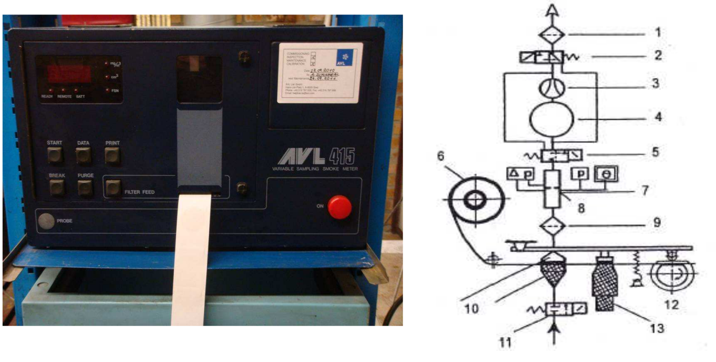


Fig. 3.40 View and diagram of the measurement system of a filtration smokemeter [5]
1- filter, 2, 5 and 11 – valves, 3 – diaphragm pump, 4 – compensatory vessel,
6 filter paper, 7- flowmeter, 8 – reducer, 9 - filter, 10 – setting mechanism,
12 – camshaft, 13 – optical measuring head

The control system of the smokemeter can process the filter smoke number (FSN) in the range of 0÷9.99 per smoke density unit (determined in g/m^3 in the range of 0÷1.0). FSN is the smoke equivalent on the Bosch scale, designated with the D_B symbol, and expressed in the Bosch scale units (B.s.u.).

Light obscuration smokemeters

Light obscuration smokemeters operate on the principle using quantitative changes in the intensity of light radiation, described by the Beer-Lambert law.

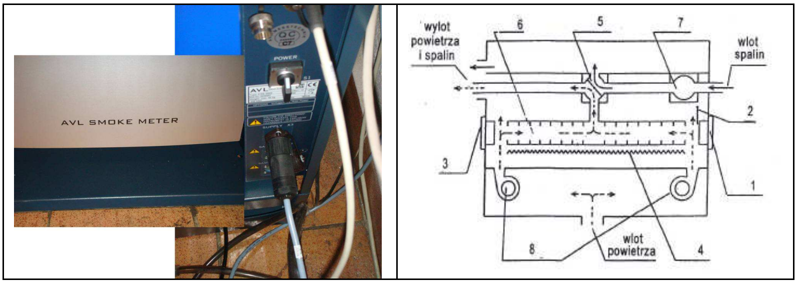


Fig. 3.41 View and functional diagram of a light obscuration smokemeter [5]
1 – halogen lamp, 2 – control diaphragm, 3 – radiation detector, 4 – heating element, 5 – switching valve, 6- measuring chamber, 7 – hot well, 8 – air fan.

Optionally, the results are displayed as the amount of clouding (opacity) (N; 0÷100 %) or light absorption (K; 0÷∞ 1/m).

Exhaust gas sampling probes

Regardless of the type of measuring device, there is a need for test sampling of gas. Shapes and dimensions of exhaust gas sampling probes as well as the sites and methods of their installation are specified by the manufacturers' instructions and/or testing standards. This issue is particularly important when gas samples are taken for identification of the degree of exhaust smoke. In such case the following parameters are important:

– probe distance from the exhaust manifold (due to the need to mix together gases from different engine cylinders),
– probe distance from pipe bends (risk of soot segregation),
– gas pressure (set flow intensity),
– gas temperature (due to the possibility of condensation of water absorbing gas components and soot particles).

Standards for emissions of toxic components

Due to the enormous diversity of applications of internal combustion engines as a power source, a lot of standards have been introduced that define the research methods and the permissible values of the respective exhaust gas components. In addition, those standards have been introduced in different regions of the world, which has also contributed to the diversity of standards and times for completion. This serves to distinguish European, American, and Japanese testing standards, but also standards of the respective countries concerning single-track vehicles, the so-called off-road vehicles, and, obviously, light and heavy vehicles, etc. Above all, the distinction concerns standards of testing spark-ignition engines in opposition to the testing of compression-ignition engines. Those standards are frequently modified due to the importance of the issue and the emergence of new engine technologies and measurement methods.

The measured values can be used to evaluate specific items of the work or to develop characteristics – chapter 5.

4. THE ENGINE THERMAL BALANCE AND EFFICIENCY

Any work related to the development of combustion engines has to take into account the discussion on thermodynamics and heat exchange, as the engine is a heat engine.

The basic tasks in this area include the thermal balance analysis. The considerations on the thermal balance of the combustion engine concern the balance between the energy supplied to the engine (Q_o) and the energy carried off in the form of effective work (L_e) and heat (Q_s) according to the following formula:

$$Q_o = L_e + Q_s$$

The value of work, L_e, defines the possibility of fulfilling the assigned task, whereas the other component part of the balance – heat, Q_s, is identified with unproductive release of energy, i.e. losses. Distinguishing the components of heat losses, Q_s, makes it possible to establish the influence of respective factors on the engine operation in various conditions and provides the basis for recognising two types of balance: external and internal one.

4.1. External thermal balance of the engine

This balance is prepared on the basis of measurements of mechanical work and thermal energy given up by the engine to the outside. Such balance is determined under the conditions of the steady thermal balance of the engine.

The basic thermal balance equation, derived from the law of conservation of energy, takes the following form in the case of the external balance [11, 28]

$$Q_o = L_e + Q_{ch} + Q_w + Q_{ns} + Q_{dys} + Q_r$$

where:

- Q_o - the total amount of heat supplied to the engine,
- L_e - effective work,
- Q_{ch} - heat carried off to the cooling medium,
- Q_w - exhaust heat losses,
- Q_{ns} - the amount of heat lost due to incomplete fuel combustion,
- Q_{dys} - dissociation heat losses,
- Q_r - the rest of the balance, containing indefinable radiation heat losses,

Expressing the parameters of the above equation as a function of time it is possible to obtain the equation of thermal fluxes, i.e. power, which is more convenient for technical calculations:

$$\dot{Q}_o = P_e + \dot{Q}_{ch} + \dot{Q}_w + \dot{Q}_{ns} + \dot{Q}_{dys} + \dot{Q}_r$$

In order to create the balance it is necessary to measure or calculate the values of its respective components. The thermal flux supplied to the engine Q_o is determined according to the following formula:

$$\dot{Q}_o = G_e \bullet Q_{HV}$$

where:

- G_e - weight of fuel delivered over a unit of time,
- Q_{HV} - fuel calorific value,

To determine this parameter the calorific value of the used fuel has to be known, which, for example, for diesel oil amounts to Q_{HV} = 42,700 kJ/kg, and for petrol Q_{HV} = 43,120 kJ/kg [11, 26]. The weight of the fuel consumed over a unit of time, G_e, has to be measured at the test stand, employing the gravimetric or volumetric method - Section 3.

The first component of the balance, N_e, represents the value of effective power of the engine, which should be determined at the test stand by way of measuring the torque – Section 3.

The second component is the thermal flux related to cooling medium losses, Q_{ch}. In general, this flux consists of the sum of thermal fluxes carried off from the engine itself, its systems and elements. It can be, for example, the thermal flux carried off by the water cooler or oil cooler. It can also be the radiant heat of the cylinder block.

Depending on the type of cooling, creating the balance of the discussed losses can be more or less complicated. It can be achieved in the simplest way in the engines with liquid cooling, i.e. equipped with the so-called indirect cooling systems.

In this case the tested engine should be mounted in the test stand equipped with two measurements systems: one making it possible to measure the stream of the cooling liquid and its temperature at the inflow and outflow of the engine and the other making it possible to measure the stream of the cooling liquid and its temperature at the inflow and outflow of the oil crankcase cooling system. In this case the thermal flux attributable to radiation can be determined according to the following dependence:

$$\dot{Q}_{ch} = \dot{Q}_{ch1} + \dot{Q}_{ch2} + \dot{Q}_{p}$$

where:

Q_{ch} - thermal flux losses attributable to cooling
Q_{ch1} - thermal flux losses attributable to cooling of the cylinder block
Q_{ch2} - thermal flux losses attributable to cooling of the oil crankcase
Q_p - thermal flux losses attributable to radiation of the cylinder block.

In many cases the last component of the equation, i.e. the thermal flux of the cylinder block radiation, Q_p, is omitted in these considerations due to the difficulties related to its determination. Thus the value of Q_p increases the value of the rest of thermal flux losses, represented in the general equation as Q_r. The remaining components of the sum are determined according to the following dependence:

$$\dot{Q}_{ch1} = \dot{m}_1 \bullet c_{w1} \bullet \Delta T_1$$

$$\dot{Q}_{ch2} = \dot{m}_2 \, c_{w2} \bullet \Delta T_2$$

where:

m_1, m_2 - cooling medium streams
c_{w1}, c_{w2} - specific heat of the cooling liquid
ΔT_1, ΔT_2 - difference in the temperatures between the inflow and outflow of the cooling system indexes
$_1$ – cooling system of the cylinder block
$_2$ – cooling system of the oil sump.

Assuming that the influence of Q_p is negligible and that the specific heat of the cooling liquid in both systems is identical, we can arrive at the following equation:

$$\dot{Q}_{ch} = c_w (\dot{m}_1 \Delta T_1 + \dot{m}_2 \Delta T_2)$$

The measurement of the stream of liquid in the cooling systems is performed by filling the tank up to the specified volume or by means of flowmeters incorporated in the cooling systems. In both cases it is necessary to measure the time and to take account of the dependence of the density of the liquid on the temperature. In the case of direct cooling systems (surplus heat is carried off directly to the environment) or mixed systems the procedure for determination of the cooling losses is more complex. First of all it requires the extension of the test stand by adding a blower with the controlled air flow and temperature and thermal shields ensuring the tightness of the measurements system.

Another component of the balance is the exhaust thermal flux losses marked as Q_w. Exhaust heat losses result from removal of exhaust gases from the engine, and the temperature and pressure of such exhaust gases are higher than the ambient temperature and pressure.

The determination of the Q_w thermal flux consists in measuring the stream of gases and their temperature in the exhaust system and establishing their specific heat.

$$\dot{Q}_w = \dot{n}_s (Mc_p)_s T_s - \dot{n}_p (Mc_p)_p T_p$$

where:

- n - number of kilomoles of gas per a unit of time,
- (Mc_p) - mean specific heat at constant pressure,
- T - temperature,
- indexes: s – applicable to exhaust gases,
  p – applicable to air.

The number of kilomoles of exhaust gases per a unit of time is determined according to the following formula:

$$\dot{n}_s = \frac{i\,n}{120} n_s$$

where:

- i - number of cylinders,
- n - engine rotational speed,
- n_s - number of kilomoles of exhaust gas.

The dynamic character of the outflow of exhaust gas, combined with the stoichiometric changes occurring over time, makes the measurement of the number of kilomoles of exhaust gas, n_s, an extremely difficult procedure. Therefore, in order to determine n_s an indirect – computational - method is employed, where the number of kilomoles of sucked air, n_p, is established and a specific value of the molar enthalpy change factor, μ, is adopted from the range of $\mu = 1.02 \div 1.04$.

$$n_s = n_p\,\mu$$

where:

- n_p - number of kilomoles of sucked air,
- μ - molar enthalpy change factor ($\mu = 1.02 \div 1.04$)

An error committed in this procedure arises mainly from the error related to adopting a specific value of μ and it is estimated at the level of a few percent.

In the case of a laboratory procedure for determination of the number of kilomoles of sucked air, n_p, established on the basis of the following formula, it is necessary to measure the ambient temperature and pressure and the engine capacity and to determine the volumetric efficiency, η_v, for the specific conditions of the balance, according to the method described in Section 3.

$$n_p = \frac{p_{ot}}{T_{ot}} \frac{V_s}{MR} \eta_v$$

where:

p_{ot} - ambient pressure,
T_{ot} - ambient temperature,
V_s - engine capacity,
MR - universal gas constant (MR = 8314,3 J/(kmol K),
η_v - volumetric efficiency.

For the mean specific heat of exhaust gas $(Mc_p)_s$ it is necessary to make an analysis of the composition of exhaust gas in the specified conditions of the engine operation, and then to establish the shares of respective components of exhaust gas. The specific heat $(Mc_p)_s$ at constant pressure is derived from the following formula:

$$(Mc_p)_s = (Mc_v)_s + MR$$

where:

$(Mc_p)_s$ – molar specific heat at constant pressure,
MR – universal gas constant,
$(Mc_v)_s$ - molar specific heat at constant volume

The value $(Mc_v)_s$ is defined on the basis of the following dependence:

$$(Mc_v)_s = (Mc_v)_{N_2}\Big|_t^0 \cdot z''_{N_2} + (Mc_v)_{O_2}\Big|_t^0 \cdot z''_{O_2} +$$
$$+ (Mc_v)_{CO_2}\Big|_t^0 \cdot z''_{CO_2} + (Mc_v)_{H_2O}\Big|_t^0 \cdot z''_{H_2O}$$

where:

z'' - shares of respective components,
(Mc_v) - molar specific heat at constant volume in the temperature range of 0 ÷ t °C (taken from the thermodynamic tables)

The molar specific heat of air $(Mc_p)_p$ at constant pressure is determined in the same way as the molar specific heat of exhaust gas, taking into account the shares of respective components in the air.

Two subsequent components of the thermal balance, i.e. heat losses caused by incomplete fuel combustion, Q_{ns}, and dissociation heat losses, Q_{dys}, should be considered jointly, making use of the following dependence, and they are referred to as combustion losses.

$$\dot{Q}_{ns} + \dot{Q}_{dys} = (1 - \xi)\, G_e\, Q_{HV}$$

where:

G_e – hourly fuel consumption
Q_{HV} – fuel calorific value
ξ – heat release rate

The heat release rate, ξ, is a measure of the thermal load of the combustion chamber. The procedure for its determination requires the knowledge of molar enthalpy changes and stoichiometry of the working medium. Under the laboratory conditions the values of this rate are taken from the tables, defined on the basis of the engine type, the range of engine's rotational speed and the combustion chamber type. For the nominal rating of the engine operation the values of ξ fall into the following ranges [11, 22, 33]:

- compression-ignition engines	$\xi = 0{,}70 \div 0{,}90$
- supercharged compression-ignition engines	$\xi = 0{,}60 \div 0{,}80$
- spark-ignition engines	$\xi = 0{,}85 \div 0{,}95$
- gas engines	$\xi = 0{,}80 \div 0{,}85$

The last component of the external balance is the thermal flux loss attributable to radiation, called the rest of the balance, Q_r. It is formed, for example, by already mentioned radiant heat, resulting from the heating up of the engine elements caused by exhaust gas or cooling water, part of the heat carried off to the outside, which originated as a result of friction, and also the heat equivalent to the kinetic energy of exhaust gas.

The thermal balance of the engine can be well illustrated in a graphic form, i.e. the so-called Sankey diagram. An example of such diagram has been shown in the figure below.

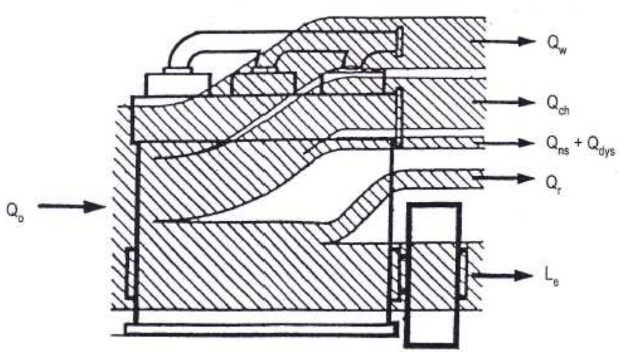


Fig.4.1 Sankey diagram presenting the engine thermal balance [28]

Taking into account the dependence of respective components of the balance as a function of engine rotational speed or mean effective pressure, it is possible to present the engine thermal balance in the variable operating conditions, as shown in the figure below.

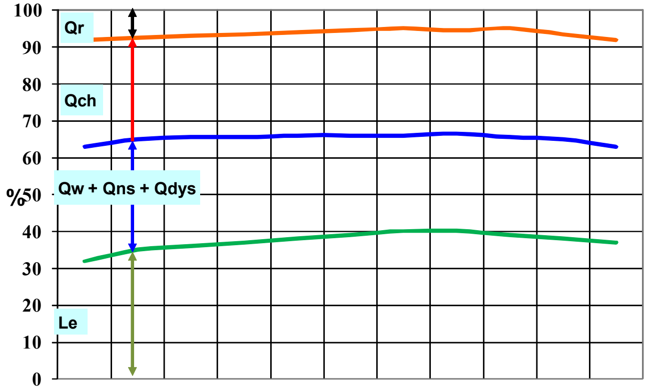


Fig. 4.2 Thermal balance of the diesel engine as a function of mean effective pressure

4.2. Internal thermal balance of the engine

The internal thermal balance of the combustion engine defines the heat distribution in the cylinder, and therefore in the literature it can be encountered under a different name – the thermal balance of a cylinder. This type of balance makes it possible to assess in detail the engine operation and its design and to determine the influence of various operating factors. The basis for the proper creation of the internal balance is provided by an accurately plotted indicator diagram – Section 3. Thence, the availability of appropriate measurement instruments is a necessity. The fundamental difference, in comparison with the external balance, arises from the fact that it makes use of indicated work (L_i) instead of effective work (L_e), or, in principle, of a thermal flux converted to this work (Q_i). The Q_i heat combines the heat converted into effective work, Q_e, and the heat losses resulting from resistance to motion, Q_m. The Q_i thermal flux is calculated on the basis of the pressure time-course in the combustion chamber. The internal balance takes also into account the thermal flux losses attributable to cooling and exhaust heat losses. Thus, the internal balance can be described by means of the following mathematical dependence:

$$\dot{Q}_o = \dot{Q}_e + \dot{Q}_m + \dot{Q}_{ch} + \dot{Q}_w$$

where: Q_o - thermal flux supplied to the engine,
Q_e - effective thermal flux,
Q_m - thermal flux attributable to resistance to motion,
Q_{ch} – heat losses attributable to cooling,
Q_w - exhaust heat losses.

Exhaust heat losses, Q_w, include the losses resulting from incomplete combustion. The basic formula for the balance may be subject to numerous modifications, depending on the conducted tests, related to the accurate definition of the distribution of heat losses attributable to cooling. Thence, it is necessary to know the geometrical data of the engine elements and the information about the materials they are made of. Furthermore, it is necessary to install thermometer probes measuring the temperature and thermal fluxes in the specified areas.

Such tests are very important from the point of view of assessing the correctness of the combustion process and thermal loads of the elements surrounding the combustion chamber and of other elements, directly related to them. Most often the creation of the internal balance is accompanied by the determination of the losses attributable to cooling of pistons, cylinder sleeve, cylinder head, exhaust passages, turbocharger, lubricating oil and supercharged air.

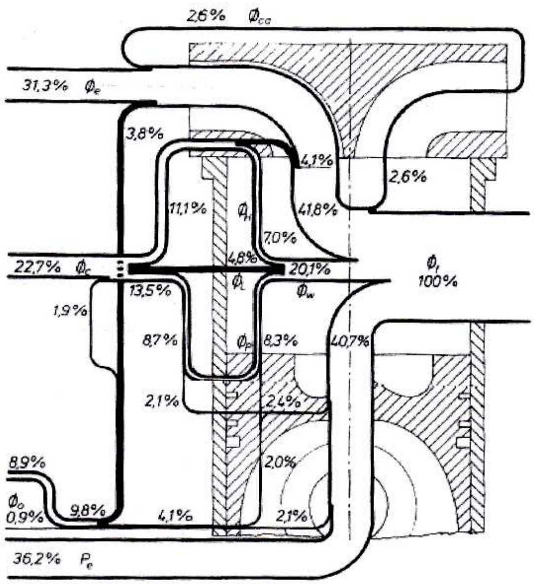


Fig. 4.3 Sankey diagram illustrating the internal balance of a turbocharged diesel engine, Valmet 611CS (V_{ss} = 6.6 dm3, D=108 mm, S=120 mm, n = 1,400 rpm, p_e=1.04 MPa, Q_o=37.8 kW) [17]

4.3 Engine efficiency

For the purpose of assessing the design of a combustion engine and the energy changes taking place in it, the term 'efficiency' is used. Depending on the assessed phenomenon occurring in the combustion piston engine the following efficiencies can be distinguished:

- ideal efficiency (reference engine cycle) η_t,
- indicated efficiency η_i,
- thermal efficiency η_c,
- mechanical efficiency η_m,
- general efficiency (effective) η_e.

All of them have been described in Section 2. It is worth mentioning that the effective efficiency can be determined in a relatively simple way under the laboratory conditions, provided that the measurements of effective power and hourly fuel consumption are taken into account and the fuel calorific value is known, using the following formula:

$$\eta_e = \frac{P_e}{\dot{Q}_o} = \frac{P_e}{G_e \bullet Q_{HV}}$$

Taking into consideration the dependencies discussed so far, the thermal balance of the engine can be also expressed as the so-called balance of shares.

$$1 = 1 - \xi + \eta_e + q_{ch} + q_w + q_r$$

where:

ξ - heat release rate,
η_e – effective engine efficiency,
$q_{ch} = Q_{ch}/G_eQ_{HV}$ - share of cooling losses,
$q_w = Q_w/G_eQ_{HV}$ - share of exhaust losses,
$q_r = Q_r/G_eQ_{HV}$ - share of the remaining losses.

Indicative values of the effective engine efficiency for different engines have been listed below:

– compression-ignition engines $\eta_e = 0.27 \div 0.38$
– supercharged compression-ignition engines $\eta_e = 0.35 \div 0.45$
– spark-ignition engines $\eta_e = 0.22 \div 0.35$
– gas engines $\eta_e = 0.36 \div 0.40$

5. COMBUSTION ENGINE CHARACTERISTICS

The combustion engine performance is the relation between such parameters of the engine operation as power, torque, fuel consumption and rotational speed. The same term applies to the dependence of the parameters of the engine operation on the factors affecting such operation, e.g. the influence of the ignition advance angle on the engine power or the influence of the ambient temperature on the hourly fuel consumption.

The engine characteristics make it possible to assess the technical condition or the results of retrofitting and to compare different types of engines.

Characteristics are most often presented in a graphic form.

Typical engine characteristics can be divided into:

– speed characteristics,
– load characteristics,
– regulation characteristics.

Speed characteristics express the dependence of the parameters of the engine operation on its rotational speed. The most common speed characteristics include:

– wide-open-throttle operating characteristics (for ships the so-called propeller characteristics),
– partial power characteristics,
– smoke limit characteristics,
– performance map.

Load characteristics present the relationship between various parameters of the engine operation and its load.

Regulation characteristics describe the relationship between the typical parameters of the engine operation and changes in the settings of the systems controlling the engine operation, for example:

- fuel injection advance angle characteristics, which assumes the existence of the relationship between the engine power and various values of the fuel injection advance angle,
- ignition advance angle characteristics, covering the issue of identification of the relationship between the parameters of the engine operation and the ignition start,
- fuel blend composition characteristics, which is used to describe the dependence, for example, of the engine torque on the fuel consumption, resulting in the combustion process which is total and complete.

5.1 Wide-open-throttle operating characteristic

The wide-open-throttle operating characteristic is the most popular one among all characteristics and it takes its name from the determination of the area of the engine operation in the conditions where there is a maximum fuel charge to compression-ignition engines or a full throttle in spark-ignition engines.

In the literature references a historical name of this characteristic can be found, related to the engine tests during its natural operation, i.e. mounted in the vehicle. The idiom 'step on it' or 'put the pedal to the metal' (in Polish 'gaz do dechy' – meaning 'put the pedal to the wood plank') is related to the maximum fuel charge, obtained in the vehicle by pressing the pedal to the floor. In former times the floor was made of wood, commonly called 'decha' in Polish (wood plank). This old name is still used in the sports environment and among people involved in tuning.

In accordance with the standards the wide-open-throttle operating characteristic can be determined for the so-called 'net' or 'gross' engine equipment – as shown in the tables below. Thus, power or torque as well as other parameters of the engine operation are referred to using these expressions. Therefore, the net power will be the power achieved at the crankshaft (or its equivalent) by the engine with specific equipment, with a full throttle in the case of the spark-ignition engine or with an extreme position of the lever of the device controlling the fuel charge, ensuring its maximum value, in the case of the compression-ignition engine.

Table 5.1 Engine equipment for the tests of net parameters

No.	Name of the equipment	Engine equipment for determination of the net power (torque)
1	2	3
1	Inlet system - inlet manifold - ventilation system of the crankcase - air filter - air induction silencer - high-speed governor	 standard equipment standard equipment[1)]
2	Heater (glow plug) for fuel blend or for sucked air	standard equipment, set at the most favourable position as indicated by the manufacturer

3	Exhaust system - afterburner for exhaust gas - exhaust manifold - supercharging device - connecting exhaust pipes[2)] - exhaust silencer[2)] - exhaust silencer outlet pipe[2)] - exhaust brake[3)]	standard equipment, the brake should be fully open or disassembled
4	Fuel supply pump[4)]	standard equipment
5	Carburettor - electronic control system, flowmeter, etc. (if assembled) - pressure regulator - vaporizer - mixer	standard equipment, regulation – according to the manufacturer's data for the standard equipment engine equipment for engines fed with gas fuel
6	Fuel injection system - filters - fuel injection pump - high pressure pipes - fuel injectors - throttle of the inlet system controlling the vacuum speed governor, if assembled[5)] - electronic control system, flowmeter (if assembled) - speed governor	standard equipment, regulation – according to the manufacturer's data for the standard model
7	Liquid (water) cooling system - radiator - fan[6),7)] - fan guard - water pump - thermostat[8)]	standard equipment[6)], radiator shutter should be open, detachable fan should be switched on
8	Air cooling system - engine sheet metal covers - blower[6),7)] - temperature controller	standard equipment, detachable blower should be switched on
9	Electrical equipment	standard equipment[9)], the load attached to the generator should come only from the electric equipment related to the engine operation (including the electric cooling fan), it is inadmissible to use the generator to charge the battery, setting of the ignition – according to the ignition advance angle characteristics provided by the manufacturer for the standard model
10	Supercharging device - supercharger or turbosupercharger - air cooler[10)] - cooling medium pump or fan - cooling medium flow regulator	standard equipment
11	Ancillary fan at the test stand	if necessary
12	Device for reducing exhaust gas toxicity[11)]	standard equipment

1)	A standard inlet system should be used in the following cases: - when a different inlet system could have a noticeable impact on the engine power, - two-stroke spark-ignition engines, - upon manufacturer's request. In other cases an equivalent system could be used, however the suction pressure should not differ by more than 100 Pa from the limit pressure defined by the manufacturer for the clean air filter.
2)	A standard exhaust system should be used in the following cases: - when a different exhaust system could have a noticeable impact on the engine power, - two-stroke spark-ignition engines, - upon manufacturer's request. In other cases an equivalent system could be used, however the pressure at the outlet of the engine exhaust system, 0.15 m behind the exhaust system, should not differ by more than 1000 Pa from the pressure defined by the manufacturer.
3)	If the exhaust brake is incorporated in the engine, the throttle should be set in the fully open position.
4)	The fuel pressure in the fuel supply system can be regulated, if necessary, in order to recreate the pressure occurring in the given engine system, in particular when the fuel recirculation system is used.
5)	The speed governor may include other devices influencing the amount of the injected fuel.
6)	The radiator, the fan, the guard, the water pump and the thermostat should be situated at the test stand in the same way as they are located in the vehicle. The coolant circulation should be driven by the engine water pump only. Liquid cooling can take place either in the engine radiator or in the external circuit, provided that the pressure losses in such circuit and the pressure at the pump inlet correspond to the pressure in the engine cooling system. If the fan, the radiator and the guard cannot be properly attached to the engine, it is necessary to determine the power input of the fan separately mounted in a proper position with regard to the radiator and the guard. The power should be obtained on the basis of the characteristics for the speed corresponding to the engine speed during the measurements of the net power or in practical tests.
7)	A detachable fan should be disconnected during the measurements, whereas the progressive fan should operate with the maximum slip.
8)	The thermostat should be set in the fully open position.
9)	The minimum power of the generator should enable the operation of all equipment indispensable for the engine operation. If it is necessary to connect the battery, it should be fully loaded and in a good technical condition.
10)	Engines which are supercharged with cooled air should be tested together with the liquid or air cooling system, however at the engine manufacturer's request the test stand system can be replaced with the air cooler. In any case the power measurement at each rotational speed should be conducted with the same drop in the pressure and engine temperature, via the supercharged air cooler or via the test stand system, as observed in a fully equipped vehicle and defined by the manufacturer.
11)	The device may consist, for example, of an EGR system, a catalytic converter, a thermal reactor, an additional air supply system and a system protecting against fuel vaporisation.

Table 5.2 Engine equipment for the tests of gross parameters

No.	Name of the equipment	Engine equipment for determination of the gross (power) torque
1	Inlet system - inlet manifold - air filter - air induction silencer - ventilation system of the crankcase - speed governor	 standard equipment standard equipment or not to be used – to be decided by the manufacturer not to be used
2	Heater (glow plug) for fuel blend or for sucked air	standard equipment, in the case of manual control preheating should be set for the summer season
3	Exhaust system - afterburner for exhaust gas - exhaust manifold - connecting exhaust pipes - exhaust silencer - exhaust silencer outlet pipe - exhaust brake	 standard equipment or not to be used – to be decided by the manufacturer test stand equipment[1)] not to be used
4	Fuel supply pump	standard equipment
5	Carburettor	standard equipment, regulation – can be set at the maximum power according to the manufacturer's data
6	Fuel injection system - filters - fuel injection pump - high pressure pipes - fuel injectors - throttle of the inlet system controlling the vacuum speed governor, - speed governor	 standard equipment or the test stand equipment standard equipment, regulation – it can be set for the maximum power according to the manufacturer's data standard equipment, the throttle should be fully open standard equipment, reset for the maximum rotational speed or disassembled
7	Liquid (water) cooling system - radiator - fan - fan guard - water pump - thermostat	 not to be used standard equipment, the thermostat can be set in the fully open position
8	Air cooling system - engine sheet metal covers - blower[2)] - temperature controller	standard equipment, the cooling system should be in the fully open position not to be used
9	Electric equipment	standard equipment, if necessary, the load attached to the generator should come only from the electric equipment indispensable for the engine operation, it is inadmissible to use the generator to charge the battery, the ignition can be set according to the ignition advance angle characteristics for the maximum power specified by the manufacturer

10	Supercharging device - supercharger or turbosupercharger - air cooler[3)] - cooling medium pump or fan[2)] - cooling medium flow regulator	standard equipment not to be used
11	Device for reducing exhaust gas toxicity	standard equipment (normal setting) or not to be used – to be decided by the manufacturer

1) The system for taking exhaust gas off the test stand should not generate, at the outlet of the exhaust manifold, the pressure that would differ from the ambient pressure by more than 740 Pa, unless the manufacturer has consented to such increased counterpressure.

2) The engine power should be increased by adding the power input of the blower (fan), determined at the same rotational speeds.

3) The supercharged air cooler may be replaced with another cooler, provided that the supercharging pressure and the supercharging temperature are the same as those defined by the manufacturer for the engine supercharging system.

Apart from power, torque and hourly fuel consumption, the wide-open-throttle operating characteristic may also express other dependencies, related, for example, to the change in the content of toxic components of the exhaust gas accompanying the change in the engine rotational speed.

For the external characteristic the following rotational speeds can be distinguished:

– n_o – the lowest rotation speed at which the engine can operate with the external load, overcoming the resistance to motion and internal friction,
– n_{max} – the maximum rotational speed at which the engine can operate without any disturbance.
– n_N – the rotational speed at which the power reaches its nominal value,
– n_M – the rotational speed at which the engine torque reaches its maximum value,
– n_g – the rotational speed at which the specific fuel consumption, g_e, reaches its minimum value.

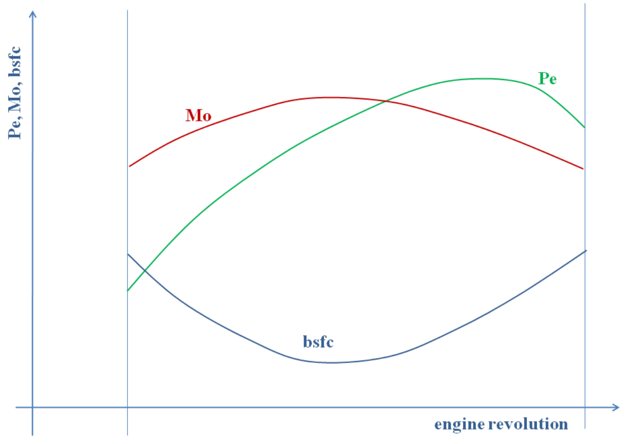


Fig. 5.1 Wide-open-throttle operating characteristic of a combustion engine

When the wide-open-throttle operating characteristic is determined, the parameters used to define the engine response are measured – Section 2.

While discussing the wide-open-throttle operating characteristic there should be mentioned the divergence of power curves (the phenomenon commonly referred to as engine overspeeding) in the areas of nominal speeds of spark-ignition and compression ignition engines.

In the case of the spark-ignition engine, upon excessive rotation the engine is only partially filled with the air-fuel mixture, due to a significant increase in the resistance of flow. Such resistance is directly proportional to the square of the flow speed value, and thus to the engine rotational speed. When the cylinder is filled with only a partial, and gradually decreasing, amount of the mixture, the amount of energy supplied to the engine also decreases, which results in a considerable reduction in power. In the case of the compression-ignition engine, the excessive rotation during the engine operation causes different phenomena. Also in this case the resistance of flow increases, but not to such extent as in the spark-ignition engine. It results from the fact that only air is sucked and there is always an excess of air in the compression-ignition engine as compared to the fuel dose. During the overspeeding period the power increases and to prevent it the compression-ignition engines are equipped with a speed governor. A defective governor may be the reason of major engine damage due to the excessive increase in stress caused by centrifugal forces.

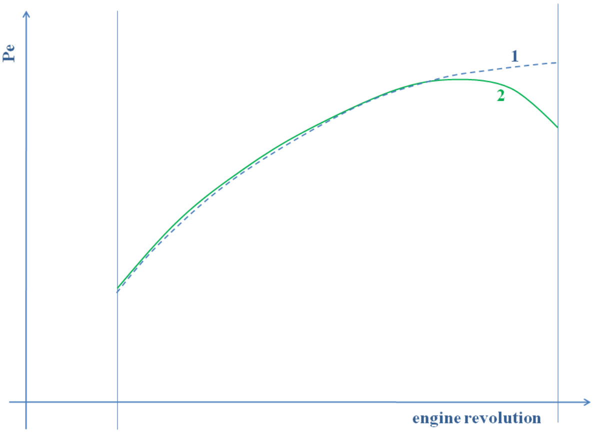


Fig. 5.2 Comparison of wide-open-throttle operating characteristics of spark-ignition (2) and compression-ignition (1) engines

5.2 Partial power characteristic

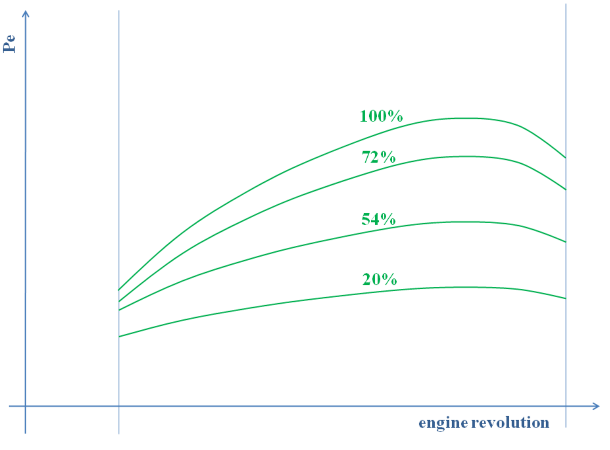


Fig. 5.3 Partial power characteristic for the spark-ignition engine

The partial power characteristic is prepared for different throttle openings for the spark-ignition engine and for different settings of the fuel injection pump for the compression-ignition engine. Figures 5.3 and 5.4 present the partial power characteristics for both engine types. 100%, 72%, 54% and 20% power curves correspond to different throttle openings. The total power, 100%, corresponds to the full opening, and the partial power to the incomplete opening of the throttle. The partial power characteristic for the compression-ignition engine has been determined with the aid of the multi-range controller.

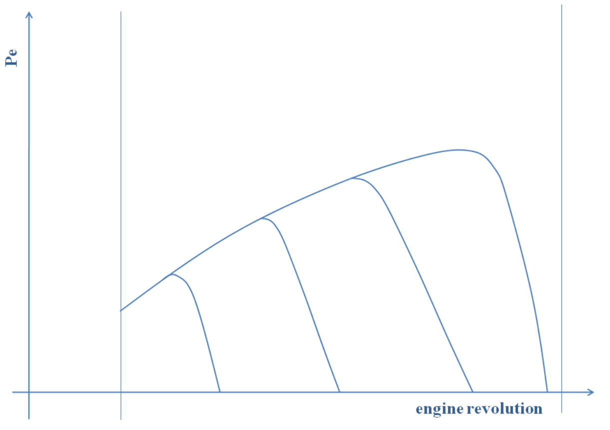


Fig. 5.4 Partial power characteristic for the diesel engine with a multi-range controller

5.3 Smoke limit characteristic

The smoke limit characteristic most often presents the mutual dependence of power and mean effective pressure as a function of rotational speed at the settings of the devices controlling the fuel dose corresponding to the smoke limit for each rotational speed.

The smoke limit characteristics are prepared for the compression-ignition engines.

By supplying to the combustion chamber a fuel dose higher than the dose corresponding to the wide-open-throttle operating characteristic, a certain increase in power is obtained, however it is done at the expense of worse combustion and increased fuel consumption. The symptom of this phenomenon is a change in the colouring of the exhaust gas into a grey and black colour, and hence, its name is derived from the commonly used term 'smoke'.

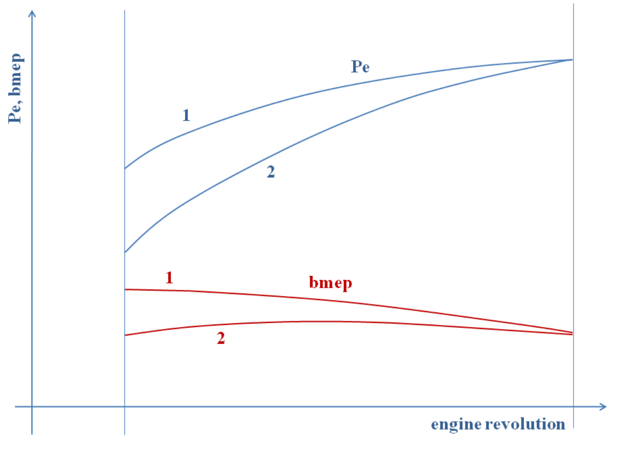


Fig. 5.5 Smoke limit characteristic.
1 – smoke limit, 2 – wide-open-throttle operating characteristic

The smoke limit corresponds to the power curve, which is higher than the power curve of the wide-open-throttle operating characteristic whose line is usually rather flat. Smoke occurs also because of the excess air number values decreasing with the increase in speed. When approaching the maximum power, corresponding to high rotational speeds, the wide-open-throttle operating characteristic and the smoke characteristic are brought nearer to each other – as shown in the figure 5.5.

5.4 Propeller characteristic

The propeller characteristic presents the dependence of the selected parameters of the combustion engine operation on the rotational speed when the engine cooperates with the ship's screw propeller. The propeller characteristic usually includes the curves of specific and fuel consumption per hour and the mean effective pressure curve as well as the calculated curve of effective power input.

The determination of effective power is based on the assumption that the power input into the screw propeller changes in accordance with the following equation:

$$P_e = C \bullet n^3$$

where: C – a constant, depending on the design parameters of the screw propeller.
n – engine revolution

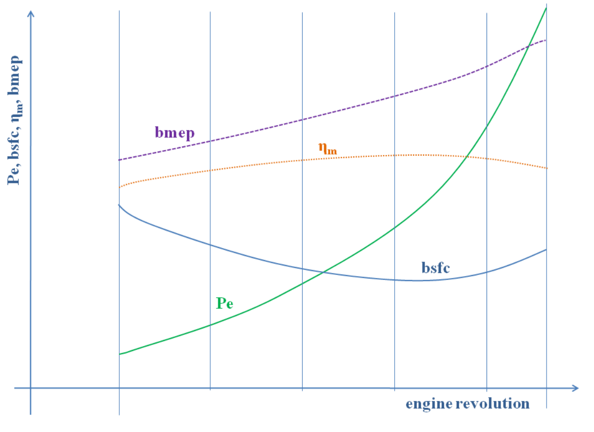


Fig. 5.6 Propeller characteristic of the compression-ignition engine

5.5 Load characteristics

A typical load characteristic of the combustion engine presents the dependence of the specific and/or fuel consumption per hour as a function of effective power or effective pressure, measured at the constant rotational speed.

The load characteristic corresponds to the constant rotational speed, and hence to the selected vertical section of the partial power characteristics.

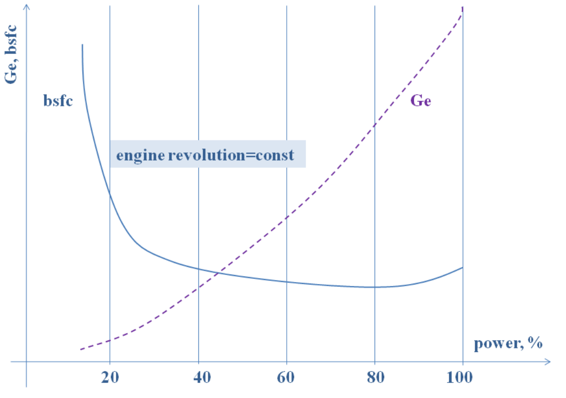


Fig. 5.7 Load characteristic of the compression-ignition engine

5.6 Regulation characteristics

Regulation characteristics describe the dependencies of the selected parameters of the engine operation on the regulated factor, for example the fuel injection advance angle. On the basis of the obtained results it is possible to undertake development or regulation activities, e.g. code the central system controlling the fuel dose in order to avoid reaching the smoke limit or optimise the blend composition for ecological purposes.

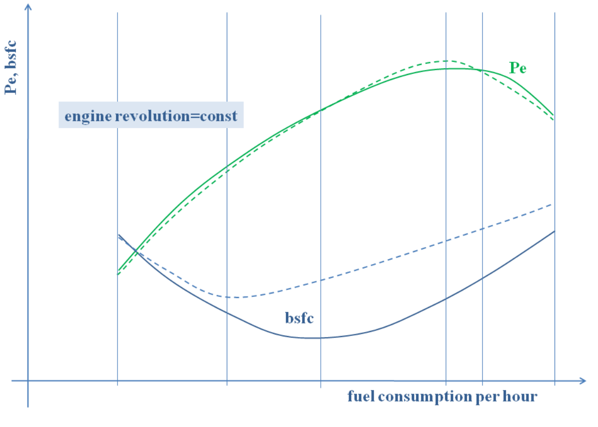


Fig. 5.8 Regulation characteristic
Pe – useful power, bsfc – brake specific fuel consumption

5.7 Performance map

The universal characteristic describes the dependence of any of the operating parameters on the load and rotational speed. It means that there is a specific three-dimensional chart of dependencies. The evaluated parameter (most often the specific fuel consumption) is presented in the form of contour lines – constant curves. The load (on the axis of ordinates) can be expressed by the torque or the mean effective pressure, and the speed is presented on the axis of abscissae – as shown in the figure below. Also the constant power lines are marked on the chart. The performance map makes it possible to analyse the engine operation parameters in the whole potential area of operation.

Instead of or next to the lines of specific fuel consumption the experimenter may insert the contour lines of other parameters, e.g. the content of toxic components of the exhaust gas or the temperature of the cooling medium. The suitability of this characteristic for making various analyses explains its addition name – “common”.

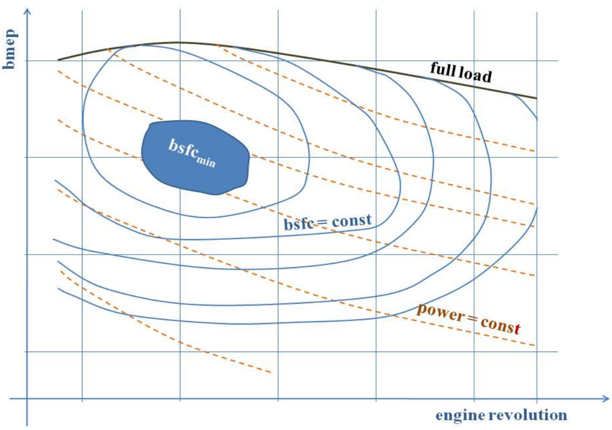


Fig. 5.9 Performance map of the compression-ignition engine

The performance map is prepared on the basis of numerous load characteristics or partial speed characteristics.

The curves of the constant specific fuel consumption or of another parameter are determined in the following way:

– a span is adopted for the layer, e.g. for g_e – it is usually 10 g/kWh,
– on the chart of the load characteristics a grid of horizontal lines is plotted, corresponding to the adopted span of the layer,
– the points where the grid lines intersect with the characteristics are plotted on the p_e – n chart, obtaining the points with constant values which, after joining them, will create the contour lines.

Hyperboles of the constant power are obtained on the basis of the following dependence:

$$P_e = V_{ss} \bullet n \bullet bmep = const$$

where with the use of the following units n [rpm], V_{ss} [dm^3], N_e [kW], bmep (brake mean effective pressure) [MPa] we can arrive at:

- for the four-stroke engine

$$bmep \bullet n = \frac{120 \bullet P_e}{V_{ss}}$$

- for the two-stroke engine

$$bmep \bullet n = \frac{60 \bullet P_e}{V_{ss}}$$

With the specific power value adopted, the above formulas can be used for calculating the mean effective pressure, which, in combination with the adopted rotational speed of the engine, gives the possibility of plotting the hyperbole of the constant power.

5.8 Power reduction to the normal conditions

The measured parameters of the engine operation are the reactions to the settings of the control systems, adopted for the tests, but they are also influenced by the environmental conditions. In order to ensure comparability, the conditions of engine tests should be identical. However, it would entail considerable financial expenditure necessary to ensure the infrastructure of the test stand totally isolated from its environment, where the tests would not be 'susceptible' to the variable values of temperature, pressure or humidity. It is considerably easier, however at the defined level of accuracy, to transform the obtained results to such values that would be obtained if the tests were conducted in the so-called normal conditions. It is referred to as the so-called reduction to the normal conditions.

In the Polish standards the normal conditions are defined as the air temperature of 25^{o}C and the atmospheric pressure of 999 hPa.

The reduction to the normal conditions is carried out by using the correction factor, K_o, which:

- for spark-ignition engines is determined as

$$K_o = \left(\frac{99}{p_o}\right)^{1,2} \bullet \left(\frac{T_{zs}}{298}\right)^{0,6}$$

where:
T_{zs} – absolute temperature of the sucked air,
p_o – ambient pressure decreased by the value of the partial pressure of water vapour, kPa

By means of the K_o factor it is possible to adjust the power, the torque, the mean effective pressure and the hourly fuel consumption in spark-ignition engines.

The specific fuel consumption is calculated by inserting into the formula the power value that has not been adjusted (!).

- for compression-ignition engines the correction factor is calculated in the following way:

$$K_d = f_a \bullet f_m$$

where:
f_a – factor taking account of atmospheric conditions,
f_m – factor taking account of the engine type and regulation

$$f_a = \frac{99}{p_o}\left(\frac{T}{298}\right)^{0,7}$$

$$f_m = 0{,}036 \cdot g_c - 1{,}14$$

where:

g_c – a fuel dose in milligrams per litre of the engine cubic capacity (mg/cycle)
for $g_c < 40$ mg - $f_m = 0.3$ is adopted
for $g_c > 65$ mg - $f_m = 1.2$ is adopted

5.9 Principles of preparing characteristics

In order to determine characteristics it is necessary to follow the basic rules of conduct, which guarantee safety at work and arrival at the correct measurement results.

Before starting the engine, first it is necessary to:

- identify the type of characteristic to be determined and the range of engine rotation and load,
- inspect the engine and the test stand to verify the correctness of installation
- of the measurement instruments and pieces of equipment, and to check the mounting of the engine and of the test stand,
- perform typical activities related to the engine maintenance such as checking the levels of lubricating oil, fuel and cooling liquid,
- check the operation of the measurement instruments,
- prepare the measurement sheet, i.e. the document in which the measured values will be entered.

The engine can be started up after the listed activities have been completed.

After starting the engine it is necessary to:

- determine the engine's rotational speed for the idle run,
- check the readings of the measurement instruments,
- heat the engine up to the operating temperature, i.e. to about 80^{o}C of the cooling medium.

Before preparing each of the characteristics it is necessary to measure the fuel consumption per hour without any load, for the idle run rotation. This measurement is conducted to check the level of engine heating and the correctness of operation of its basic systems – mainly the fuel related devices. This measurement is also the so-called closing measurement for each test series, conducted after the characteristic has been determined.

The measurements of fuel consumption, power and other tested parameters for respective points of the characteristic are conducted at the steady thermal state of the engine. It can be verified whether the thermal state is steady by observing the readings of the measurements instruments – the stability of the instrument readings evidences the thermal balance of the engine. The number of measurement points required for the preparation of the characteristic depends on its purpose and the necessary accuracy and usually amounts to 10. When preparing the wide-open-throttle operating characteristic the measurement points are set at every 200-500 rpm, however the number of points is usually made higher in the area of inflexion of the characteristic, i.e. for the maximum power and torque.

The following rules should be observed for recording of the measurement points:

- the readings of the measurement instruments should be constant for at least 1 minute,
- the engine's rotational speed should not change by more than 1% of the assumed rotation or ± 10 rpm,
- measurement should last about 30 seconds in the case of automatic recording and about 60 seconds in the case of manual recording,
- ambient temperature and pressure should be within the limits of 5-30^{o}C and 730-765 mm Hg, respectively.

After analysing the measurement results and making necessary recalculations the data for preparing the characteristic are available.

APPENDIX no. 1

Examples of error analyses

1) Calculation of the reading error of the rotational speed measurement

The rotational speed of a combustion engine is measured directly on the crankshaft by means of an instrument with the inaccuracy class of 0.5 and the measuring range of 10,000 rotations per minute.

The absolute value of the admissible error determined on the basis of the instrument class is as follows:

$$\text{class} = \frac{\Delta y_{dop} \bullet 100}{Z}$$

after rearranging

$$\Delta y_{dop} = \frac{Z \bullet \text{class}}{100} = \frac{10{,}000 \bullet 0.5}{100} = \pm \quad 50 \quad \text{rpm}$$

Relative errors

- for the indication of n = 1,000 rpm it is as follows:

$$\delta y = \pm \frac{\Delta y}{y_{\text{measured}}} \bullet 100 = \pm \frac{50}{1{,}000} \bullet 100 = \pm \quad 5\%$$

- for the indication of n = 5,000 rpm it is as follows :

$$\delta y = \pm \frac{\Delta y}{y_{\text{measured}}} \bullet 100 = \pm \frac{50}{5{,}000} \bullet 100 = \pm \quad 1\%$$

The above example shows that the measurement at the higher range of the instrument scale is burdened with a smaller relative error, therefore it is recommended to include in the measurements predominantly the ranges falling above 50% of the range.

2) Calculation of errors of the fuel consumption per hour (fuel flow) measurement

The hourly fuel consumption is a value measured indirectly and determined according to the following dependence:

$$G_e = \frac{V_m \bullet \rho_p}{\tau}$$

where:

V_m - volume of fuel consumed,
ρ_p - fuel density,
τ - time of measuring fuel consumption,

The absolute error of the determination of an indirect value Δy is established on the basis of the following formula:

$$|\Delta y| = \frac{\partial f}{\partial x_1}|\Delta x_1| + \frac{\partial f}{\partial x_2}|\Delta x_2| + ...$$

where:

- Δx_1, Δx_2, - systematic errors of respective direct measurements
- $\frac{\partial f}{\partial x_1}, \frac{\partial f}{\partial x_2}, ...$ - partial derivatives of the y function

and hence

$$\Delta G_e = |\frac{\partial G_e}{\partial V_m}|\Delta V_m + |\frac{\partial G_e}{\partial \rho_p}|\Delta \rho_p + |\frac{\partial G_e}{\partial t}|\Delta t$$

$$\Delta G_e = \pm (|\frac{\rho_p}{\tau}|\Delta V_m + |\frac{V_m}{\tau}|\Delta \rho_p + |\frac{V_m \rho_p}{\tau^2}|\Delta t)$$

for the following data

- $V_m = 50\ cm^3$
- $\Delta V_m = \pm 0.2\ cm^3$
- $\rho_p = 0.00083\ kg/cm^3$
- $\Delta \rho_p = \pm 0.00001\ kg/cm^3$
- $\tau = 107.3\ s$
- $\Delta \tau = \pm 0.1\ s$
- 3,600 – conversion factor for measurement units

$$G_e = 3{,}600 \bullet \frac{50 \bullet 8.3\ 10^{-4}}{107.3} = 1.392\ kg/h$$

The absolute error of the measurement of the fuel consumption per hour ΔG_e

$$\Delta G_e = \pm 3{,}600\ (|\frac{8.3 \bullet 10^{-4}}{107.3}| \bullet 0.2 + |\frac{50}{107.3}| \bullet 1 \bullet 10^{-5} + |\frac{50 \bullet 8.3 \bullet 10^{-4}}{(107.3)^2}| \bullet 0.1)$$

$$\Delta G_e = \pm 0.024\ \ kg/h$$

The relative error of the measurement of the hourly fuel consumption δG_e amounts to

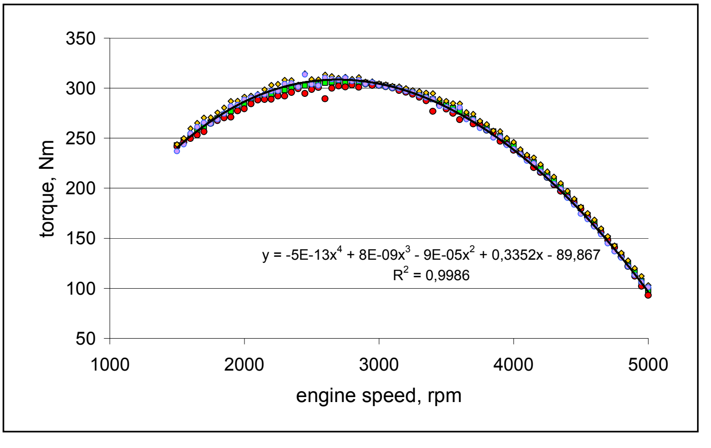
$$\delta G_e = \frac{\Delta G_e}{G_e} \bullet 100\,\%$$

$$\delta G_e = \frac{0.024}{1.392} \bullet 100\,\% = 1.724\,\% \approx 2\%$$

The biggest influence on the accuracy of the fuel consumption measurement as an indirect value is exerted by a systematic error of measurement of the consumed volume of a test fuel charge and an error of determination of the fuel density. An error of the time measurement is related to the observer's skills. If the accuracy of measurement of the test fuel charge volume was increased from 0.2 cm^3 to 0.1 cm^3 the total relative error of determination of the fuel consumption would be reduced to 1.509 %.

3) Regression function estimation taking as an example the chart of torque as a function of rotational speed of a combustion engine

The figure below presents the results of several measurement series of the torque of a combustion engine, conducted by different observers using the same object and employing the same measuring instruments, at a similar time. The recorded spread of values made it necessary to arrange the results by creating mathematical links between the input (rotational speed of the engine) and output values (torque). On the assumption that the only measurement errors were the systematic errors of the measuring instruments, it was possible to use a fourth degree polynomial for the purpose of describing the dependence, and the correlation coefficient confirmed the validity of the adopted estimation.



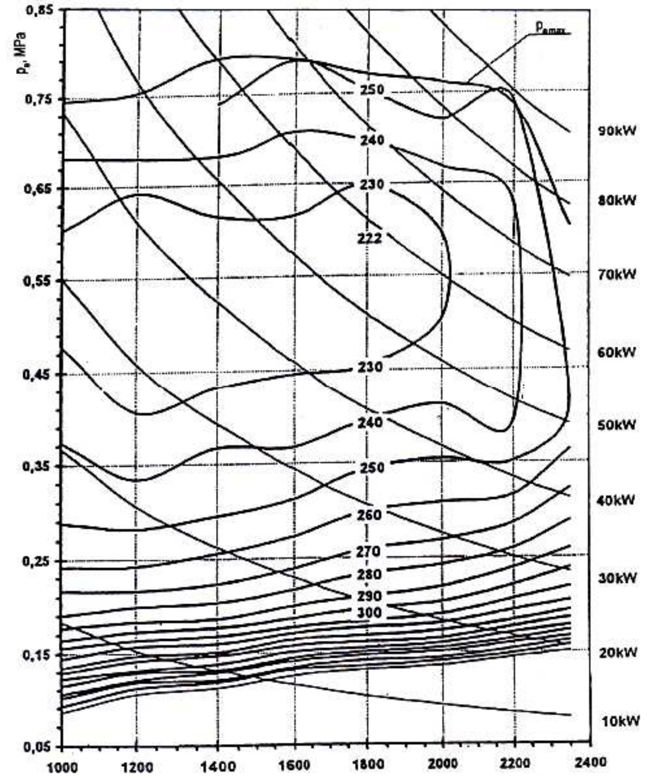
APPENDIX no. 2

Performance map determination

An example for diesel engine - SW 400 type [5]

In order to prepare the performance map of specific fuel consumption a number of load characteristics for SW 400 engine were completed, presenting the dependence $g_e = f(p_e)$ and covering the full range of effective rotational speed, from n = 1200 rpm to n = 2350 rpm, with the speed staging at every 200 rpm.

Indirect engine operating parameters were calculated following formulas given in section 2.



Performance map of the specific fuel consumption

Measurement sheet – Diesel engine type SW-400/_2

No.	n	Mo	time for fuel consumption		temperature			pressure		CO	NO_2	D	CO_2
			50 cm^3	200 cm^3	lub. oil	water	exhaust	lub. oil	exhaust				
	rpm	Nm	s	s	°C	°C	°C	MPa	mmHg	%	ppm	Bosch	%
1	2350	314		29,8	40	60	579,0	0,45	20	0,37	1668	3,79	10,2
2	2200	383		26,2	57	79	685,0	0,40	20	0,75	1646	6,01	12,6
3	2000	396		28,8	70	83	673,0	0,39	18	0,45	1790	5,67	12,6
4	1800	401		31,3	85	85	659,0	0,40	15	0,33	1834	5,77	12,4
5	1600	410		35,6	86	95	653,0	0,32	13	0,32	1856	5,93	12,6
6	1400	411		39,6	95	86	623,0	0,29	12	0,58	2030	5,81	11,0
7	1200	392		49,8	89	85	570,0	0,25	10	0,45	2345	5,54	10,6
8	1000	388		61,7	85	85	554,0	0,23	10	0,45	2260	5,11	11,0
9	1000	40	87,2		80	65	176,0	0,25	6	0,12	270	0,11	1,9
10	1000	100	51,4		75	62	204,0	0,30	6	0,07	500	0,28	3,2
11	1000	200	30,2		73	63	299,0	0,33	7	0,07	1319	1,93	5,5
12	1000	300	21,0		70	68	403,0	0,35	7	0,12	2106	3,73	7,8
13	1000	380	15,5		69	75	524,0	0,34	10	0,45	2163	4,81	10,6
14	1200	40	65,9		70	68	188,0	0,40	6	0,12	260	0,07	2,0
15	1200	100	42,1		69	61	222,0	0,42	7	0,09	473	0,26	3,2
16	1200	200	25,8		68	60	308,0	0,42	8	0,04	1167	0,77	5,2

17	1200	300	17,6		68	63	408,0	0,42	8	0,12	1725	2,80	7,6
18	1200	390	13,1		70	75	556,0	0,38	10	0,50	2086	5,02	10,6
19	1400	40	54,1		69	69	220,0	0,41	7	0,10	322	0,05	2,1
20	1400	100	35,8		69	63	232,0	0,41	7	0,11	446	0,09	3,3
21	1400	200	21,7		69	62	320,0	0,42	7	0,05	1115	0,97	5,4
22	1400	300	15,2		69	64	434,0	0,43	8	0,13	1720	2,67	7,8
23	1400	408	9,8		70	74	600,0	0,44	10	0,65	1752	6,28	11,4
24	1600	40	44,6		70	66	234	0,42	8	0,08	311	0,13	2,1
25	1600	100	30,3		70	61	250	0,42	9	0,10	413	0,16	3,4
26	1600	200	19,0		70	60	335	0,42	8	0,05	1030	0,68	5,5
27	1600	300	13,2		70	63	443	0,42	9	0,12	1600	2,25	8,0
28	1600	410	8,8		71	70	598	0,45	10	0,55	1700	6,80	11,4

Load Characteristics

No	n	Mo	Ne	pe	Ge	ge	CO	NO_x	D	CO_2
	rpm	Nm	kW	MPa	kg/h	g/kWh	%	ppm	Bosch	%
1	1000	40	4,19	0,08	1,73	412,5	0,12	270	0,11	1,9
2	1000	100	10,47	0,19	2,93	279,9	0,07	500	0,28	3,2
3	1000	200	20,94	0,38	4,99	238,2	0,07	1319	1,93	5,5
4	1000	300	31,41	0,58	7,17	228,4	0,12	2106	3,73	7,8
5	1200	380	39,79	0,73	9,72	244,3	0,45	2163	4,81	10,6
6	1200	40	5,03	0,08	2,29	454,9	0,12	260	0,07	2,0
7	1200	100	12,57	0,19	3,58	284,8	0,09	473	0,26	3,2
8	1200	200	25,13	0,38	5,84	232,4	0,04	1167	0,77	5,2
9	1200	300	37,70	0,58	8,56	227,1	0,12	1725	2,80	7,6
10	1200	390	49,01	0,75	11,50	234,7	0,50	2086	5,02	10,6
11	1400	40	5,86	0,08	2,78	474,9	0,10	322	0,05	2,1
12	1400	100	14,66	0,19	4,21	287,1	0,11	446	0,09	3,3
13	1400	200	29,32	0,38	6,94	236,8	0,05	1115	0,97	5,4
14	1400	300	43,98	0,58	9,91	225,4	0,13	1720	2,67	7,8
15	1400	408	59,81	0,78	15,37	257,0	0,65	1752	6,28	11,4
16	1600	40	6,70	0,08	3,38	504,1	0,08	311	0,13	2,1
17	1600	100	16,75	0,19	4,97	296,8	0,10	413	0,16	3,4
18	1600	200	33,51	0,38	7,93	236,6	0,05	1030	0,68	5,5
19	1600	300	50,26	0,58	11,41	227,1	0,12	1600	2,25	8,0
20	1600	410	68,69	0,79	17,12	249,2	0,55	1700	6,80	11,4
21	1800	40	7,54	0,08	4,00	530,1	0,07	300	0,14	2,5
22	1800	100	18,85	0,19	5,73	303,9	0,10	427	0,10	3,6
23	1800	200	37,70	0,38	9,13	242,2	0,05	945	0,68	5,6
24	1800	300	56,54	0,58	12,56	222,0	0,10	1445	1,81	7,9
25	1800	394	74,26	0,76	18,60	250,5	0,55	1616	5,89	11,6
26	2000	40	8,38	0,08	4,58	546,7	0,07	340	0,29	2,7
27	2000	100	20,94	0,19	6,47	308,8	0,08	462	0,16	3,8
28	2000	200	41,88	0,38	10,25	244,7	0,05	930	0,47	5,8
29	2000	300	62,83	0,58	14,35	228,4	0,09	1525	1,79	8,3
30	2000	392	82,09	0,75	20,93	254,9	0,60	1482	5,58	11,8
31	2200	40	9,21	0,08	5,29	573,7	0,07	322	0,31	2,9
32	2200	100	23,04	0,19	7,46	323,8	0,07	453	0,23	3,9
33	2200	200	46,07	0,38	11,16	242,2	0,05	932	0,52	6,0

34	2200	300	69,11	0,58	16,38	237,0	0,09	1405	1,98	8,4
35	2200	390	89,84	0,75	22,49	250,3	0,58	1460	5,46	12,2
36	2350	40	9,84	0,08	6,15	624,8	0,06	328	0,57	3,2
37	2350	100	24,61	0,19	8,19	332,7	0,07	455	0,31	4,2
38	2350	200	49,21	0,38	12,56	255,1	0,05	927	0,58	6,2
39	2350	300	73,82	0,58	17,94	243,0	0,12	1386	2,30	8,8
40	2350	333	81,94	0,64	19,82	241,9	0,22	1504	3,83	10,0

Hyperboles of constant power

n	Ne max	Ne [kW]									
		94	90	80	70	60	50	40	30	20	10
rpm	kW	pe [MPa]									
1000	94	1,72	1,65	1,47	1,28	1,10	0,92	0,73	0,55	0,37	0,18
1200		1,44	1,38	1,22	1,07	0,92	0,76	0,61	0,46	0,31	0,15
1400		1,23	1,18	1,05	0,92	0,79	0,66	0,52	0,39	0,26	0,13
1600		1,08	1,03	0,92	0,80	0,69	0,57	0,46	0,34	0,23	0,11
1800		0,96	0,92	0,82	0,71	0,61	0,51	0,41	0,31	0,20	0,10
2000		0,86	0,83	0,73	0,64	0,55	0,46	0,37	0,28	0,18	0,09
2200		0,78	0,75	0,67	0,58	0,50	0,42	0,33	0,25	0,17	0,08
2350		0,73	0,70	0,62	0,55	0,47	0,39	0,31	0,23	0,16	0,08

Performance map of specific fuel consumption

n	pe	ge, g/kWh									
rpm	MPa	230	250	270	290	310	330	350	370	390	400
		pe, MPa									
1000	0,746	0,48	0,29	0,22	0,18	0,16	0,13	0,12	0,10	0,09	0,09
1200	0,753	0,41	0,28	0,22	0,18	0,16	0,15	0,13	0,12	0,11	0,10
1400	0,790	0,43	0,29	0,22	0,19	0,17	0,15	0,14	0,13	0,12	0,11
1600	0,788	0,45	0,31	0,24	0,20	0,18	0,16	0,15	0,14	0,13	0,12
1800	0,771	0,45	0,34	0,26	0,21	0,18	0,17	0,15	0,15	0,13	0,13
2000	0,761	0,50	0,35	0,27	0,22	0,19	0,17	0,16	0,15	0,14	0,13
2200	0,736	-	0,35	0,29	0,24	0,21	0,18	0,17	0,16	0,15	0,14
2350	0,603	-	0,41	0,32	0,26	0,22	0,19	0,18	0,17	0,16	0,15

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