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Increase of efficiency of SI engine through the implementation of thermodynamic cycle with additional expansion

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Abstract. The paper presents the theoretical basis and practical implementation of the concept of a five-stroke cycle where an additional expansion of the working medium is achieved in a separate cylinder. The work, carried out at the Cracow University of Technology, constitutes a development of the idea of five-stroke engine. This paper proposes a new theoretical thermodynamic cycle that describes the processes occurring in a five-stroke engine. It has been taken into consideration that the expansion ratio of the engine is twice the compression ratio. At the next step an analysis of the thermodynamic cycle with particular emphasis on its thermal efficiency compared to efficiency of four-stroke engine has been shown. In the second part of the article selected research results of the five-stroke engine built based on the existing four-cylinder spark ignition engine have been presented. A comparison of the total efficiency obtained by the engine in the four- and five-stroke version has been provided. Similar comparisons for specific emission of the selected exhaust components have also been given.

Key words: innovative five-stroke engine cycle, Carnot Cycle, Otto cycle, five-stroke engine, additional expansion, increase of thermal efficiency of the five-stroke engine cycle.

Nomenclature

α – heat addition pressure ratio, –
ε – compression or expansion ratio, –
η – efficiency, –
κ – specific heat ratio, –
$\Delta\eta$ – increase of efficiency, %
AFR – air-to-fuel ratio, –
BDC – bottom dead centre, –
c – specific heat, J/kg·K
M – engine torque, Nm
p – pressure, Pa
Q – heat, J
rpm – revolutions per minute, –
S – entropy, J/K
T – temperature, K
V – volume, m^3

Subscripts

1,2,3,4,5 – nodal point,
5stroke – for five-stroke engine cycle,
add – added to the cycle,
Carnot – for Carnot cycle,
ch – combustion chamber of the cylinder,
cp – compression,
cyl – cylinder,
dcp – expansion,
Otto – for Otto cycle,
p – at constant pressure,
rej – rejected from the cycle,
rel. – relative,
tot – total,
V – at constant volume.

1. Introduction

One of the ways to improve the efficiency of the internal combustion engine is the implementation of a working cycle, in which the expansion of the working medium is significantly higher than the compression. In such a situation, the amount of wasted heat could be significantly reduced, which increases the thermal efficiency of the cycle. From the name of the inventor of thermodynamic cycle that describes the work of engine with higher expansion ratio it is called the Atkinson cycle. Obtaining the aforementioned effect is possible in a classical four-stroke engine with a suitable valve timing when the intake valve is open significantly longer and closes a long time after the start of the compression stroke, even 120° CA after BDC [1]. It follows that there is a possibility of applying a high geometric compression ratio while the effective compression ratio has a safe value due to knocking. On the other hand, the small volume of the combustion chamber using the entire length of the power stroke causes the engine to have a high expansion ratio, which directly affects the increase in the thermal efficiency of the theoretical cycle and the total efficiency of the engine [2]. It should be noted at this point that the increase in total efficiency is associated with a reduction

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in maximum performance resulting from a lower volumetric efficiency [3].

The development of the idea of the engine with a high expansion ratio was proposed by the Belgian engineer Gerhard Schmitz, who patented his invention under the name five-stroke engine [4]. A diagram of the five-stroke engine concept is presented in Fig. 1.

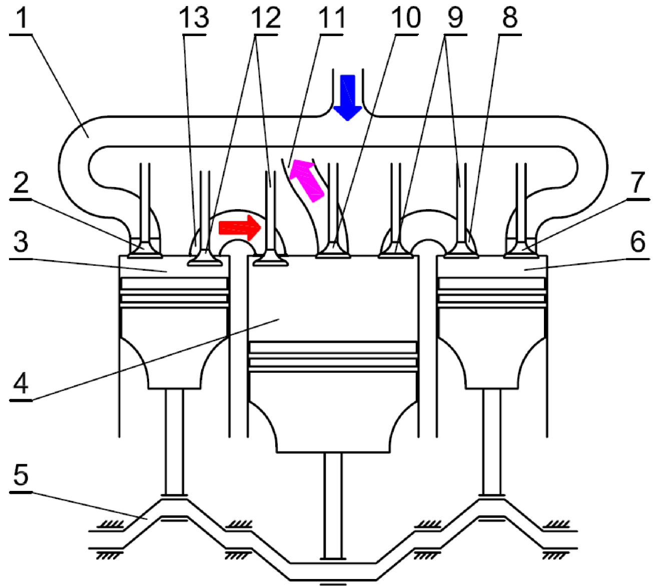

Fig. 1. Scheme of the five-stroke engine invention; 1 – Intake manifold, 2 and 7 – Intake valves, 3 and 6 – Fired cylinders, 4 – Additional-expansion cylinder, 5 – Crankshaft, 8 and 13 – Channels joining fired cylinders and additional-expansion cylinder, 9 and 12 – Valves between fired cylinders and additional-expansion cylinder, 10 – Exhaust valve, exhaust channel

The five-stroke engine process is carried out in two separate working volumes. The first of these is the fired cylinder, in which a classical four-stroke cycle is realized. The second one is a separate cylinder of about twice displacement size connected with the fired cylinder through a channel, where the exhaust gas from the first cylinder goes for further expansion. Additional expansion cylinder works in a two-stroke cycle: expansion stroke - exhaust stroke, while the fired cylinder in four-stroke cycle. Therefore, the engine has two fired cylinders. Fired cylinders work with phase shift of 360° CA relative to each other. In a result it gives realizing of the entire cycle in five strokes and explains the etymology of the name: five-stroke engine. As mentioned earlier, the volume of the working cylinder is twice lower than the volume of the additional-expansion cylinder. This combined with the work principle of the five-stroke engine make that the expansion ratio is twice as high as the compression ratio.

At the Cracow University of Technology the engine realizing the five-stroke cycle basing on in-line four-cylinder spark ignition engine was built. The idea of redesign of the four-stroke four-cylinder engine in order to implement the cycle with additional expansion of the medium is shown in Fig. 2.

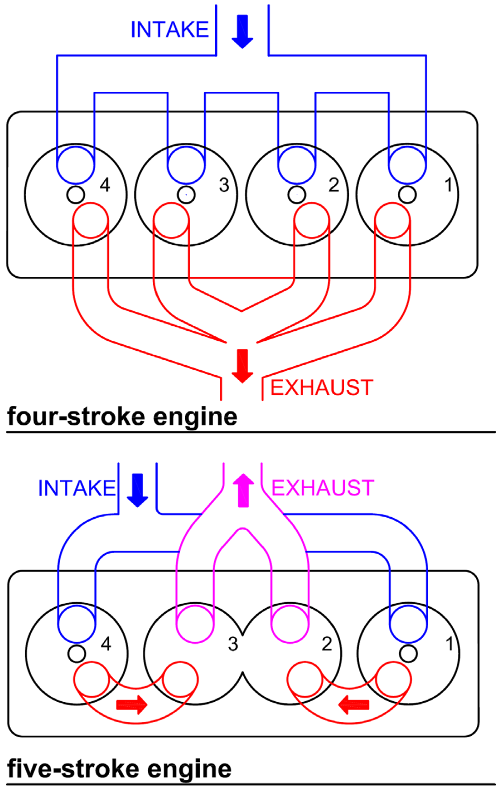

Fig. 2. Re-design of cylinder-head-flow to obtain five-stroke operation

In the redesigned engine a combustion process occurs in the cylinders No. 1 and No. 4. From the cylinder No. 1 exhaust gases go into the cylinder No. 2, while from the cylinder No. 4 into cylinder No. 3. The additional-expansion cylinders No. 2 and 3 are connected by the channel in the cylinder head. The crankshaft is classic for in-line four-cylinder engine and has all cranks in one plane arranged in the order (from cylinder No. 1) 0–180–180–360° CA. For this reason, the cylinders No. 2 and 3 work as one unit with displacement twice the displacement of the single fired cylinder. In the both cylinders No. 2 and 3 the effect of further expansion of medium is obtained, which translates into extra positive work of the cycle and improve total efficiency of the engine.

2. Thermal efficiency of the five-stroke engine cycle

In order to identify the increase in thermal efficiency of the five-stroke engine cycle of the expansion ratio two times higher than the compression ratio a theoretical comparative analysis of a four-stroke engine cycle has been conducted. In Fig. 3 a graph of the above mentioned two cycles in the p-V diagram is shown. In both cases, the same were assumed: the working medium is an ideal gas, the cylinder displacement V_{cyl}, the volume of the combustion chamber V_{ch} and the amount of heat supplied to the cycle, resulting in an isochoric increase in pressure from point 2 to 3.

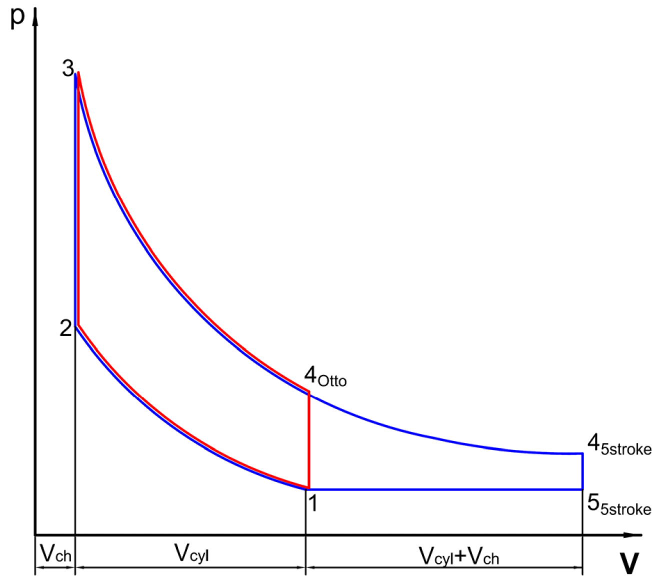

Fig. 3. Otto cycle and five-stroke engine cycle diagrams in p-V coordinates

Otto cycle of a four-stroke engine is carried out between the points $1-2-3-4_{Otto}-1$ and the five-stroke engine cycle is formed by further isentropic expansion of the medium from the point 4_{Otto} to the point $4_{5stroke}$. The volume of the working medium at the end of the expansion process (state $4_{5stroke}$) is corresponding to the total volume of both cylinders No. 2 and No. 3 of the real engine at BDC and is equal to $2(V_{cyl}+V_{ch})$. In the next stage of the five-stroke engine cycle occurs a process of heat rejection: at a constant volume, between points $4_{5stroke}-5_{5stroke}$ and at a constant pressure – $5_{5stroke}-1$.

From the perspective of further analysis, it is important to define the basic parameters characterizing the considered thermodynamic cycles. The compression ratio for both, Otto cycle, as well as for the five-stroke engine cycle is defined by the formula (1):

$$\varepsilon_{cp}=\frac{V_1}{V_2}. \tag{1}$$

The expansion ratio in the case of Otto cycle has exactly the same value as the compression ratio defined by formula (1), while for the proposed five-stroke engine cycle is described as formula (2):

$$\varepsilon_{dcp}=\frac{V_{4\_5stroke}}{V_2}. \tag{2}$$

By inserting into Eq. (2) the values of the volume as shown in Fig. 3 the following relation is yielded (3):

$$\varepsilon_{dcp}=\frac{2\left(V_{cyl}+V_{ch}\right)}{V_{ch}}=\frac{2\cdot V_1}{V_2}=2\cdot\varepsilon_{cp}. \tag{3}$$

As mentioned earlier, the expansion ratio in the five-stroke four-cylinder engine has a value two times higher than the compression ratio.

Another parameter characterizing both of the thermodynamic cycles is the heat-addition pressure ratio [5] defined by the formula (4):

$$\alpha=\frac{p_3}{p_2}. \tag{4}$$

Otto cycle thermal efficiency is given by the formula (5), which is commonly known and therefore it is not necessary to show its deriving in this paper.

$$\eta_{Otto}=1-\frac{1}{\varepsilon_{cp}^{\kappa-1}}. \tag{5}$$

For the given working medium the thermal efficiency of Otto cycle is dependent only on the assumed compression ratio [6].

To find the thermal efficiency of the five-stroke engine cycle the values of the individual components present in the general formula (6) should be determined:

$$\eta_{cycle}=\frac{Q_{add}-Q_{rej}}{Q_{add}}. \tag{6}$$

For the five-stroke engine cycle proposed by the authors of this paper, the heat is supplied at a constant volume, what is determined by the formula (7):

$$Q_{add}=m\cdot c_V\cdot(T_3-T_2). \tag{7}$$

On the other hand, the heat rejection process occurs in two stages: at constant volume and at constant pressure of the working medium – formula (8):

$$Q_{rej}=Q_{rejV}+Q_{rejp}. \tag{8}$$

The amount of heat rejected by the working fluid at constant volume is defined by the formula (9):

$$Q_{rejV}=m\cdot c_V\cdot(T_{4\_5stroke}-T_{5\_5stroke}). \tag{9}$$

Equation (10) determines the amount of heat rejected at constant pressure:

$$Q_{rejp}=m\cdot c_p\cdot(T_{5\_5stroke}-T_1). \tag{10}$$

In order to obtain a formula describing the thermal efficiency of the proposed five-stroke engine cycle the relations (7), (9) and (10) are inserted into formula (6). After making basic transformation of the formula (10) Eq. (11) is obtained:

$$\eta_{5stroke}=1-\frac{T_{4\_5stroke}-T_{5\_5stroke}}{T_3-T_2}-\kappa\cdot\frac{T_{5\_5stroke}-T_1}{T_3-T_2} \tag{11}$$

Further conversion of the formula (11) causes that it gets the form (12):

$$\eta_{5stroke}=1-\frac{T_{4\_5stroke}\cdot\left(1-\dfrac{T_{5\_5stroke}}{T_{4\_5stroke}}\right)}{T_3\cdot\left(1-\dfrac{T_2}{T_3}\right)}-\kappa\cdot\frac{T_1\cdot\left(\dfrac{T_{5\_5stroke}}{T_1}-1\right)}{T_2\cdot\left(\dfrac{T_3}{T_2}-1\right)}. \tag{12}$$

Using the equation of isentropic process 1–2, the Clapeyron equation and formula (1) the dependence (13) is obtained:

$$\frac{T_1}{T_2}=\varepsilon_{cp}^{1-\kappa}. \tag{13}$$

I the same way, a similar Eq. (14) can be written for the expression $T_{4\_5stroke}/T_3$:

$$\frac{T_{4\_5stroke}}{T_3}=(2\cdot\varepsilon_{cp})^{1-\kappa}. \tag{14}$$

A starting point to replace the expression $T_{5\_5stroke}/T_{4\_5stroke}$ is an equation of isochoric process between state 4_5*stroke* and state 5_5*stroke* (15):

$$\frac{T_{5\_5stroke}}{T_{4\_5stroke}} = \frac{p_{5\_5stroke}}{P_{4\_5stroke}}. \tag{15}$$

The pressure for point 5_5*stroke* has the same value as for point 1, thus Eq. (16) can be written:

$$\frac{T_{5\_5stroke}}{T_{4\_5stroke}} = \frac{p_1}{P_{4\_5stroke}}. \tag{16}$$

Using the equation of isentropic process 1-2 and the dependence (1), the value of pressure for state 1 can be written as follows (17):

$$p_1 = p_2 \cdot \varepsilon_{cp}^{-\kappa}. \tag{17}$$

Analogically, using the equation of isentropic process 3–4 and the dependencies (1) and (3) the pressure for state 4 is described by the formula (18):

$$p_{4\_5stroke} = p_3 \cdot (2\varepsilon_{cp})^{-\kappa}. \tag{18}$$

By inserting the dependencies (17) and (18) into Eq. (16) and using the formula (4), the dependence (19) is obtained:

$$\frac{T_{5\_5stroke}}{T_{4\_5stroke}} = \alpha^{-1} \cdot 2^{\kappa}. \tag{19}$$

In the next step, to replace the expression $T_{5\_5stroke}/T_1$ it is needed to use Eq. (20) which describes isobaric process 5_5*stroke* – 1:

$$\frac{T_{5\_5stroke}}{T_1} = \frac{V_{5\_5stroke}}{V_1}. \tag{20}$$

As it was stated before, the volume in the point 4_5*stroke* (and thus also in the point 5_5*stroke*) is two times higher than in the point 1, what means that expression (20) takes the form of (21):

$$\frac{T_{5\_5stroke}}{T_1} = 2. \tag{21}$$

The last step to give the ultimate form of equation describing thermal efficiency of the five-stroke four-cylinder engine cycle is to replace connected with each other expressions T_3/T_2 and T_2/T_3. Those dependencies are described by the equation (22) of isochoric process 2–3:

$$\frac{T_3}{T_2} = \frac{p_3}{P_2}. \tag{22}$$

It means, that the expression T_3/T_2 can be directly written as (23):

$$\frac{T_3}{T_2} = \alpha. \tag{23}$$

Analogically, the expression T_2/T_3 takes the form (24):

$$\frac{T_2}{T_3} = \alpha^{-1}. \tag{24}$$

Then, after inserting previously obtained dependencies (13), (14), (19), (21), (23) and (24) into Eq. (12) and after making basic mathematic operations, the final formula describing the thermal efficiency of the five-stroke engine cycle with the expansion ratio two times higher than the compression ratio is obtained – formula (25):

$$\eta_{5stroke} = \frac{\alpha - 1 + \varepsilon_{cp}^{1-\kappa} \cdot \left(2 - \kappa - \alpha \cdot 2^{1-\kappa}\right)}{\alpha - 1}. \tag{25}$$

In contrast to the Otto cycle, the thermal efficiency of the five-stroke engine cycle depends on both, the assumed value of the engine compression ratio ε_{cp}, as well as on the heat-addition pressure ratio α, which results from the amount of heat added to the cycle Q_{add}.

Presentation of the two considered thermodynamic cycles in the T-S diagram allows to show better the differences in the heat rejection process. This was possible by using the ideal gas law and the equations describing thermodynamic processes taking place in the both cycles.

Figure 4 shows Otto and five-stroke engine cycles in the T-S diagram, with respect to the theoretical Carnot cycle as an ideal heat engine cycle. The shown Carnot cycle is built based on maximum and minimum values of temperature the same as for Otto cycle and five-stroke engine cycle.

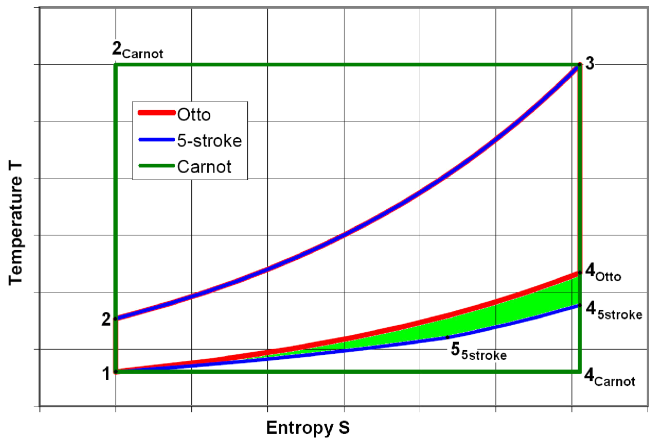

Fig. 4. Comparison of T-S diagram for Otto cycle, five-stroke engine cycle and Carnot cycle as the ideal-engine cycle

A field marked in light green colour represents the reduction in the amount of rejected heat in case of five-stroke engine cycle in relation to Otto cycle, what is directly reflected in the increase in thermal efficiency.

3. Analysis of the increase in thermal efficiency of the five-stroke engine cycle compared to Otto cycle

For the analysis the thermal efficiency of the five-stroke engine cycle the assumptions summarized in Table 1 have been made. The working medium was the air as an ideal gas. The compression ratio and the amount of heat added to the cycle were the independent variables.

Table 1
Values of parameters used for the analysis of thermal efficiency of the five-stroke engine cycle

Parameter	Symbol	Unit	Value
Specific heat at constant pressure	c_p	[kJ/kg·K]	1.005
Specific heat at constant volume	c_V	[kJ/kg·K]	0.718
Isentropic exponent	κ	[–]	1.4
Initial pressure (ambient)	p_1	[Pa]	$1.0\cdot 10^5$
Initial temperature (ambient)	T_1	[K]	300
Cylinder displacement	V_{cyl}	[m^3]	$0.496\cdot 10^{-3}$
Compression ratio	ε_{cp}	[–]	7–14
Amount of heat added to the cycle	Q_{add}	[J]	700–1400

Previous considerations resulted that the thermal efficiency of the five stroke engine cycle is dependent on value of the heat-addition pressure ratio α, which is directly proportional to the amount of heat added Q_{add} to the cycle. For the purpose of theoretical considerations a variable value of amount of heat added to the cycle and the variable value of the compression ratio were assumed, so in the first step the impact of the dependence of these quantities on the value of the heat-addition pressure ratio α was determined, as it is presented in Fig. 5.

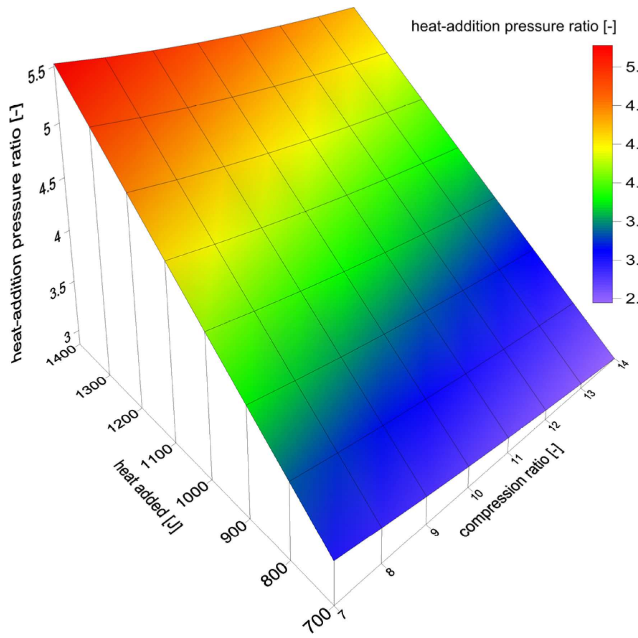

Fig. 5. Heat-addition pressure ratio α for five-stroke engine cycle in the function of heat added to the cycle Q_{add} and compression ratio ε_{cp}

Analysis of the content of Fig. 5 shows that the heat-addition pressure ratio rises with the amount of heat added to cycle and decreases with an increase in the compression ratio.

Subsequently an analysis of Otto-cycle thermal efficiency was performed. Although the thermal efficiency of the Otto cycle is not dependent on the amount of heat added to the cycle, but to preserve a uniform paper layout, the results are shown in Fig. 6 in the 3-dimensional graph: as a function of the compression ratio and the amount of heat added to the cycle.

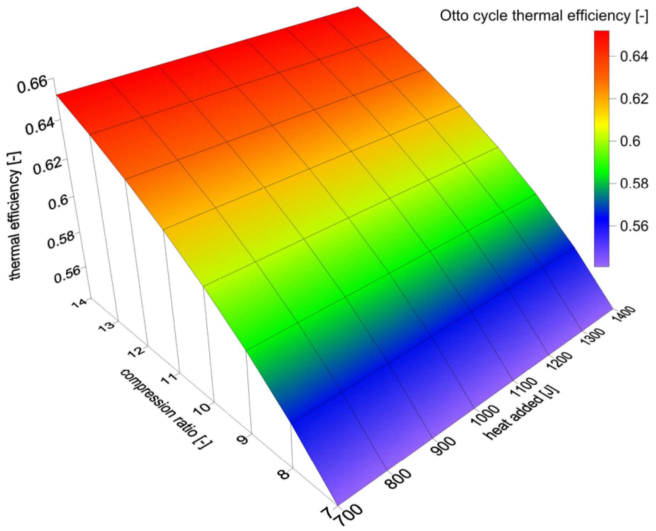

Fig. 6. Thermal efficiency of Otto cycle in dependence of heat added to the cycle Q_{add} and compression ratio ε_{cp}

Using the formula (25) the thermal efficiency of the five-stroke engine cycle was analyzed. After this analysis a three-dimensional graph showing the thermal efficiency value of the cycle depending on the amount of added heat and the compression ratio of the medium was prepared, which is shown in Fig. 7.

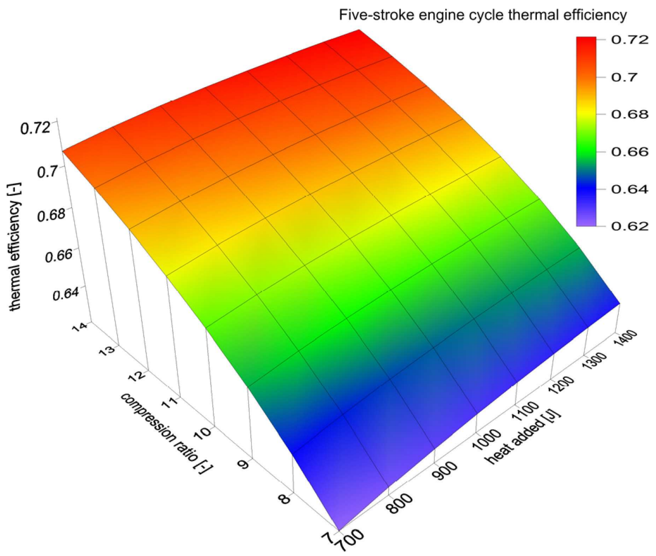

Fig. 7. Thermal efficiency of the five-stroke engine cycle in a function of heat added to the cycle Q_{add} and compression ratio ε_{cp}

As one can see in Fig. 7, the thermal efficiency of the five-stroke engine cycle increases with an increase in both the compression ratio and the amount of heat added to the cycle. The confrontation of the results presented in Figs. 6 and 7 allowed to conduct a comparative analysis of thermal efficiency of the Otto cycle and the five-stroke engine cycle. Figure 8 shows a graph of relative increase in thermal efficiency of the five-stroke engine cycle in relation to the Otto cycle, depending on the assumed value of the compression ratio and the amount of heat added to the cycle.

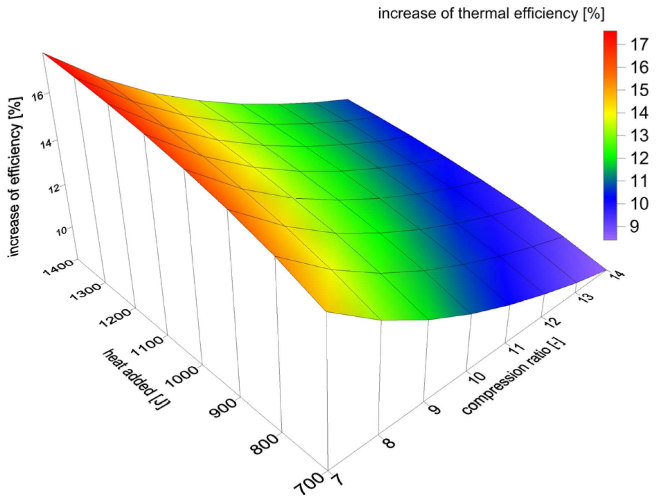

Fig. 8. Relative increase in thermal efficiency of the five-stroke engine cycle in a comparison to Otto cycle as a function of heat added to the cycle Q_{add} and compression ratio ε_{cp}

The relative increase in the thermal efficiency of the five-stroke engine cycle in relation to the Otto cycle rises with increasing the amount of heat added to the cycle. On the other hand, the dependence of the relative increase in thermal efficiency on the assumed value of the compression ratio is decreasing. For values of the compression ratio encountered in the currently manufactured spark-ignition engines, a relative increase of thermal efficiency using five-stroke cycle obtains a value between approximately 10 to 13 [%].

4. Experimental research of the five-stroke four-cylinder engine

Obtaining promising results of theoretical considerations of the thermal efficiency of the five stroke-engine cycle led that at the Institute of Automobiles and Internal Combustion Engines of Cracow University of Technology the task of developing of an engine working in accordance with the concept shown in Fig. 1 has been undertaken. The four-cylinder four-stroke engine was rebuilt according to the idea of Fig. 2 into the five-stroke engine. As an object of the study a turbocharged SI engine EA113 Volkswagen family was selected. This engine has a displacement of 1.984 [dm^3] and is equipped with a system of direct fuel injection system (TFSI). Basic technical data of the test engine are presented in Table 2, while more detailed information regarding modifications to the engine can be found in paper [7].

The above described engine was mounted on a test bed in the Laboratory of Internal Combustion Engines at Cracow University of Technology. A general view of the engine mounted on a test bench is shown in Fig. 9.

Table 2
Basic technical data of the test-engine

Parameter	Measure	Value
Displacement	dm^3	1.984
Bore × Stroke	mm	82.5 × 92.8
Compression ratio	–	10.5
Nominal output	kW at rpm	147 at 5500–6000
Maximum torque	Nm at rpm	280 at 1800–4700

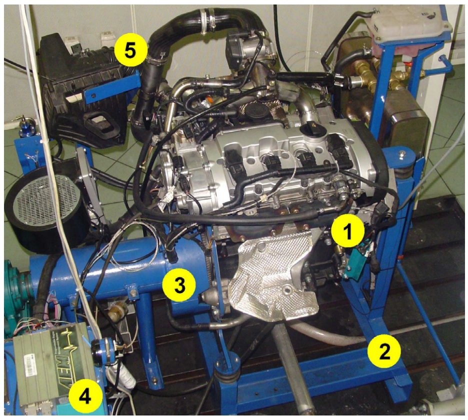

Fig. 9. Overall view of the engine test-stand; 1 – engine, 2 – test-stand frame, 3 – shaft cover, 4 – engine management module, 5 – intake system

The first series of the tests was conducted on a four-stroke engine. Then the engine was modified to work in a five-stroke cycle. Similarly, as in the original version, also after the modifications in order to implement the five-stroke cycle, the engine was equipped with a turbocharger to utilize better the remaining energy of exhaust gases [8]. Due to the fact that, for the five-stroke engine version, the actual displacement has been reduced by half, the turbocharger was adjusted accordingly. This allowed to develop a comparison of selected parameters of both engines. Figure 10 shows a comparison of the total efficiency curves obtained for both engines at variable engine load, constant rotational speed 2400 rpm and for a stoichiometric mixture composition.

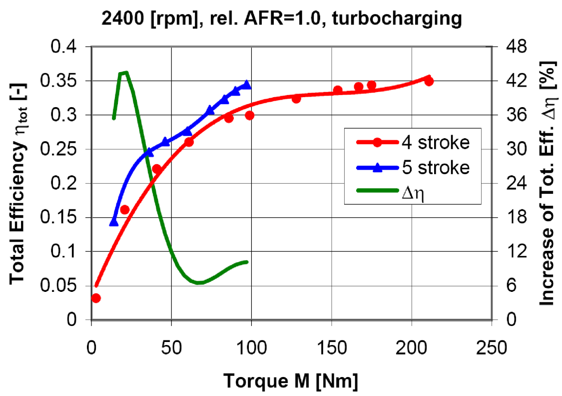

Fig. 10. Comparison of the total efficiency of the engine in four- and five-stroke version working at the rotational speed of 2400 rpm

Figure 10 shows an increase in the total efficiency of the five-stroke engine at comparable values of torque. A green line in the figure shows the course of the relative increase in the total efficiency for the five-stroke engine. The average value of the increase in the total efficiency of the five-stroke engine is 17%.

Research of emission of the selected components of the exhaust gas was also carried out. Figure 11 shows the results of the specific CO_2 and CO emissions as a function of the engine torque at the rotational speed of 2400 rpm.

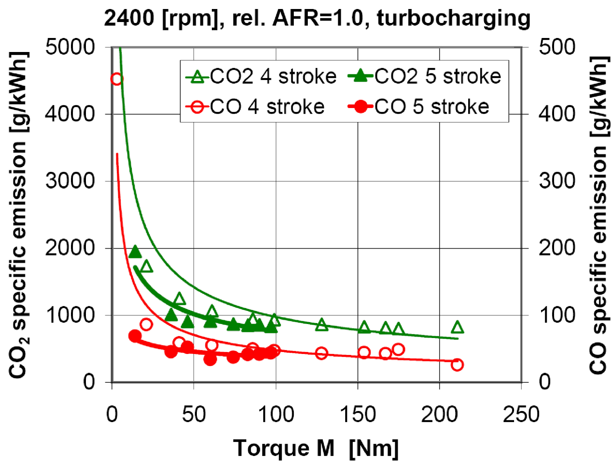

Fig. 11. Comparison of the specific emission of CO_2 and CO of the four-stroke and five-stroke engine as a function of the torque obtained at the rotational speed of 2400 rpm

Figure 11 also shows a noticeable decrease in the specific emission of CO_2 and CO in the case of the five-stroke engine. This fact is directly related to the increase in the total efficiency of the five-stroke engine under these conditions. Particularly high reduction in emissions of CO_2 and CO occurs at low engine loads.

Figure 12 shows a comparison of the results of the specific emissions of nitrogen oxide and hydrocarbons as a function of the engine torque for the test engine in four-stroke and five-stroke version recorded at 2400 rpm.

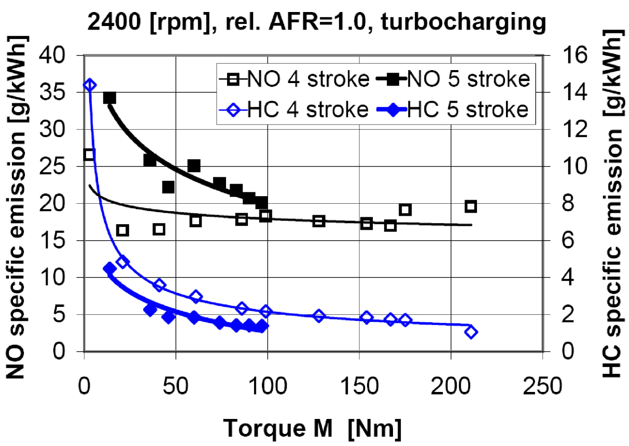

Fig. 12. Comparison of the specific emission of NO and HC of the four-stroke and five-stroke engine as a function of the torque obtained at the rotational speed of 2400 rpm

Specific emission of unburned hydrocarbons achieved lower values for the five-stroke engine in the entire range of the conducted comparison. Emissions of nitrogen oxide of the five-stroke engine is higher than for the four-stroke engine, but with increasing the load decreases so rapidly that at nominal load has a value comparable to that obtained by the four-stroke engine version.

5. Conclusions

The results of the conducted theoretical analysis and experimental research pointed out that:

1. Thermal efficiency of the five-stroke cycle of spark-ignition engine where expansion ratio two times higher than compression ratio achieved significantly higher values than it is the case of the classical four-stroke cycle (Otto).
2. The results of the theoretical analysis indicated that the relative increase in thermal efficiency of the five-stroke engine cycle in relation to the Otto cycle has an increasing trend with increasing of the amount of heat added to the cycle.
3. The research of the test-engine highlighted the reduction of CO, CO_2 and HC specific emissions, what is directly associated with the increase in total efficiency of the five-stroke engine resulting from reduced fuel consumption,
4. The increase of the total efficiency of the test-engine realizing the five-stroke cycle is so significant that the authors plan to carry out further theoretical and experimental research in this field.

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