

SIMULATION MODEL OF ADVANCED SYSTEM FOR APPLICATION OF SUSTAINABLE FUELS

Puskar, M.; Kopas, M.; Soltesova, M. & Tarbajovsky, P.

Technical University in Kosice, Letná 9, 040 01 Košice, Slovak Republic

E-Mail: michal.puskar@tuke.sk, melichar.kopas@tuke.sk, marieta.soltesova@tuke.sk,
pavol.tarbajovsky@tuke.sk

Abstract

On the present, a considerable reduction of the greenhouse gas emissions is one of the most important challenges in construction of the motorcars and their driving units. Application of the biofuels essentially contributes to reduction of air pollution caused by the exhaust gases. However, application of the bioethanol as a fuel in the spark ignition engines requires to perform some important constructional adjustments of these engines in order to ensure their proper functioning and to eliminate the potential operational risks. The scientific publication is focused on creation of a simulation model describing the advanced system, which is developed for application of the sustainable fuels. The scientific contribution of this work consists in original system of the cooling channel, which is applicable in all the high-performance sport applications. There were obtained very good results during testing of this system in real operation, especially in the case of the E85 fuel. Combustion of this fuel, using the above-mentioned system, significantly eliminated thermal loading of the whole drive unit. The authors of this scientific publication also obtained a patent for this system.

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Key Words: Simulation Model, Advanced System, Sustainability, Biofuel, Combustion

1. INTRODUCTION

The time interval, which is at disposal for removal of the combustion products (i.e. exhaust gases) from the engine cylinder and also for filling of this cylinder by a fresh fuel-air mixture is very short in the case of high powerful engines installed in the racing vehicles of the category L [1, 2]. The exchange of cylinder content is doing in the moment when the piston is in the bottom dead centre. A standard construction of the filling and exhaust pipes, which is typical for a wide range of the high powerful 2-stroke engines, is characterised by usage of five filling and four exhaust pipes (channels) [3]. There are two main disadvantages of such arrangement: insufficient filling of the cylinder with the fresh mixture and excessive thermal expansivity of the piston in cylinder. These negative phenomena are able to cause detonation combustion as well as possible destruction of engine in the case of the biofuel E85 combustion [4-8]. The main principle of the new system consists in installation of a cooling channel into the exhaust pipe and this added channel supplies the fresh fuel-air mixture into the cylinder. The cross-section of the given cooling channel is rectangular. During movement of the piston towards the bottom dead centre, the fresh mixture is transported to the cylinder volume also from the exhaust area by means of the cooling channel, which cools the exhaust part of the piston. This part of the piston is exposed to the highest temperature and this fact represents the most serious risk of destruction. This innovative constructional concept is illustrated on Fig. 1. The aim of the work is to present a system, which significantly reduces risk of the detonation combustion and it enables to use the sustainable fuels also in the racing vehicles with two-stroke engines.

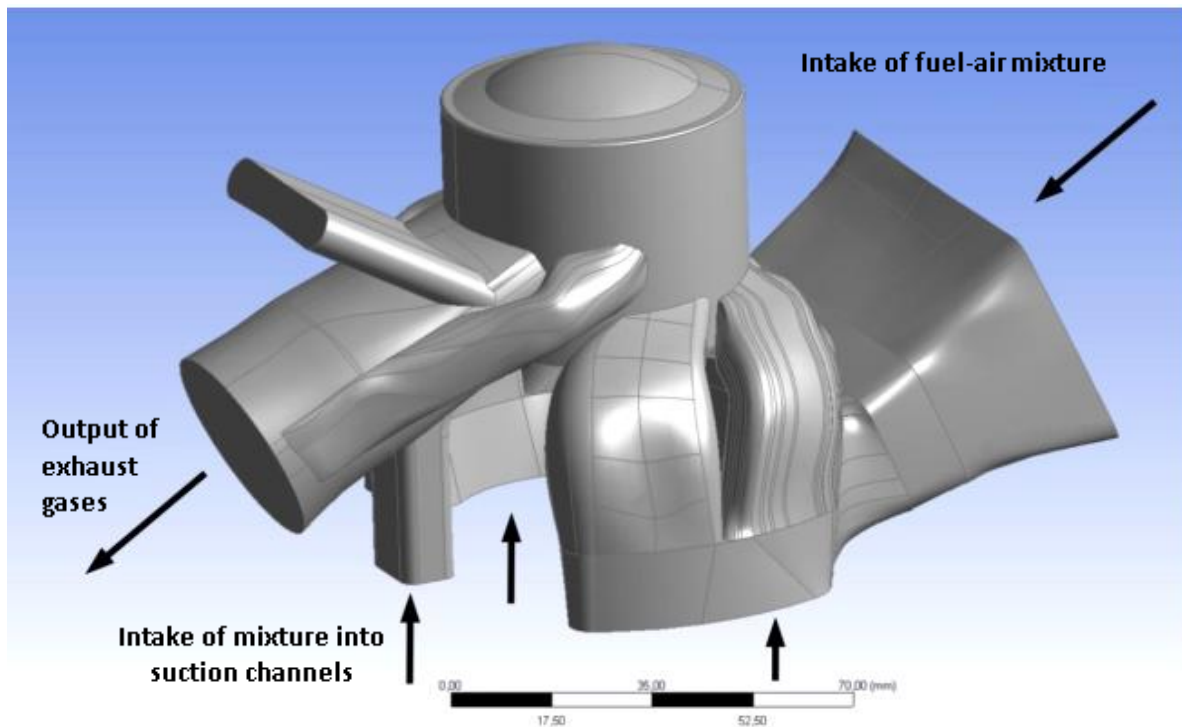


Figure 1: Inverted model of cylinder inner space with suction and exhaust channels.

2. COMPUTATION DESCRIBING EXCHANGE OF EXHAUST GASES AND FUEL MIXTURE

The most difficult problem, which occurs during a mathematical description of the Otto cycle in the case of two-stroke engine, is a complexity of gas flow and heat exchange. It is necessary to apply a suitable numerical tool in order to describe in a complex way behaviour of the basic state values during the piston movement. The required numerical tool utilizes discretisation of the cylinder model. The whole solution process was realised by means of the finite volume method.

The computational solution was based on application of the simulation software ANSYS CFX [9, 10]. This software tool enables to simulate the energy flows in the individual elements, i.e. streaming, which is described by means of the Navier-Stokes differential equations, together with complemented models of turbulence. The process of computation takes into consideration change of the network shape caused due to movement of piston. Fig. 1 presents the inverted model of the cylinder inner space of the tested engine with the suction and exhaust pipes, including the patented cooling channel. This model was obtained by a scanning procedure using the equipment METROTOM.

The inverted model, which was created by the scanning procedure, can be further used (after an adjustment) directly in the simulation software, which requires just only geometry of the inner space of cylinder. The model was simplified in order to accelerate the whole computational process, whereby the cylinder inner space remained unchanged. There were only shortened the input and output channels. Fig. 2 illustrates the final model in the moment of exhaust channels opening. Opening of the exhaust channels begins at the shift of piston 25.2 mm from the top dead centre, i.e. for the angular rotation of crankshaft at the angle $\theta_4 = 91.6^\circ$. There are set up the initialisation conditions at this value of angle, corresponding to the zero starting time $\tau = 0$ s. Consequently, increasing of the angle θ leads to the bottom dead centre (Fig. 3) and the piston starts to move again towards closing of the exhaust channels at the angle value $\theta = 268.4^\circ$.

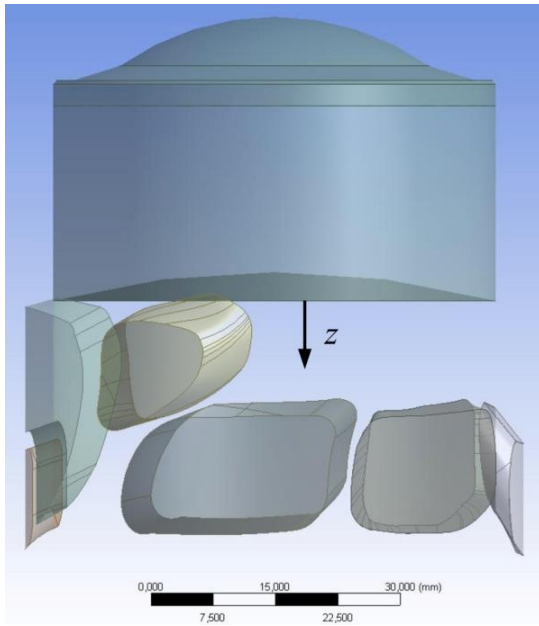


Figure 2: Investigated geometry of cylinder at $\theta_4 = 91.6^\circ$.

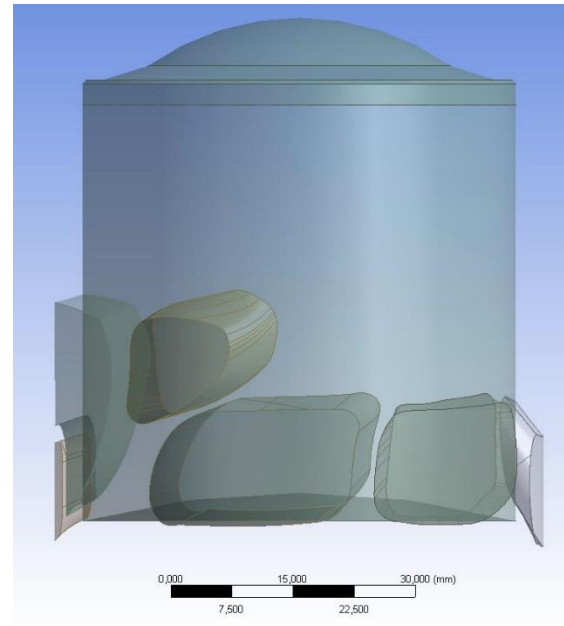


Figure 3: Investigated geometry of cylinder at bottom dead centre.

The engine speed during the simulation process was 11000 rpm. A relative shift of the piston in direction of the axis z (positive direction of the axis is defined according to Fig. 2) is given by the following relation, which describes a dependence between the simulation time and piston relative shift:

$$z = -z_0 + a \cdot \left(1 - \cos \theta - \frac{R_c}{2} \cdot \sin^2 \theta \right) \quad (\text{mm}) \quad (1)$$

where: z – shift of the piston (mm),

z_0 – position of the piston at the beginning of simulation (mm).

Time dependence of the crankshaft angular rotation at the given engine speed is described by the relation:

$$\theta = \frac{2 \cdot \pi \cdot n}{60} \cdot \tau + \theta_4 \quad (\text{rad}) \quad (2)$$

There is obtained, using the Eqs. (1) and (2), the resulting equation, which describes shift of the piston in dependence on time. This mathematical description is directly used in the simulation program:

$$z = -z_0 + a \cdot \left[1 - \cos \left(\frac{2 \cdot \pi \cdot n}{60} \cdot \tau + \theta_4 \right) - \frac{R_c}{2} \cdot \sin^2 \left(\frac{2 \cdot \pi \cdot n}{60} \cdot \tau + \theta_4 \right) \right] \quad (\text{rad}) \quad (3)$$

$$z = -25,2 + 27,25 \cdot \left[1 - \cos \left(\frac{2 \cdot \pi \cdot 11 \cdot 10^3}{60} \cdot \tau + 1,609 \right) - \frac{0,2271}{2} \cdot \sin^2 \left(\frac{2 \cdot \pi \cdot 11 \cdot 10^3}{60} \cdot \tau + 1,609 \right) \right]$$

Gradual opening and closing of the exhaust and suction channels is realised during turning of the shaft about the angle 176.8° (3.0857 rad). The time interval, which is necessary for the given angular rotation, represents duration of the simulation process. This time can be obtained from the basic equation, which describes the angular speed, using the difference form of this equation:

$$\Delta \tau = \frac{30 \cdot \Delta \theta}{\pi \cdot n} = 2,679 \cdot 10^{-3} \quad (\text{s}) \quad (4)$$

Time dependence of piston shift between opening and closing of the exhaust channels is illustrated in Fig. 4.

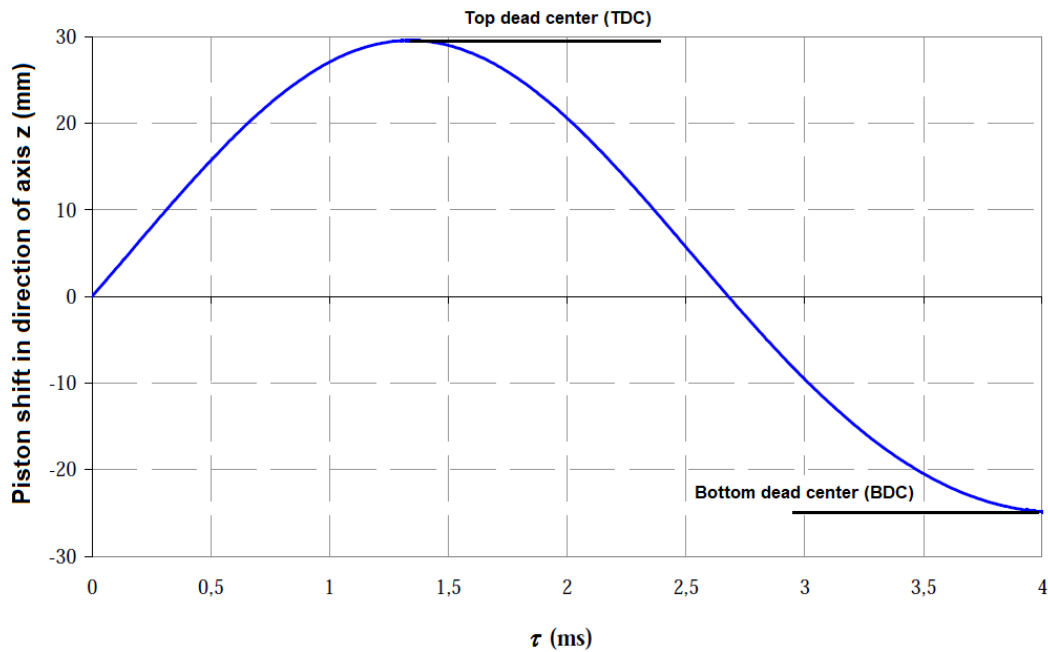


Figure 4: Time dependence of piston shift at engine speed $n = 11000 \text{ min}^{-1}$.

The model of cylinder is divided into several individual domains using the software ANSYS, namely in the part Design Modeler [11, 12]. The main reason for such fragmentation is a requirement of shifting movement concerning the bottom part of cylinder volume.

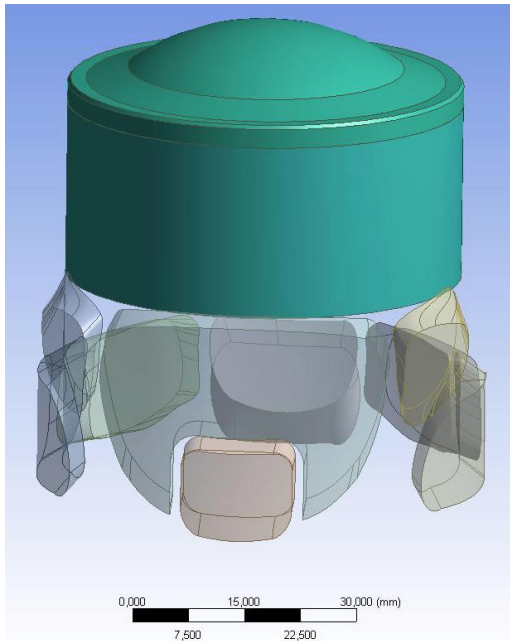


Figure 5: Inner space of cylinder at θ_4 .

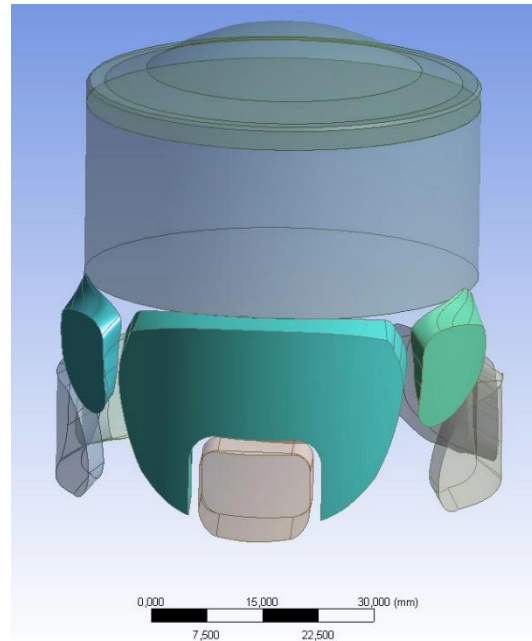


Figure 6: Exhaust channels.

The individual domains together create one functional entity. The basic computational domain is inner volume of the cylinder (Fig. 5). The exhaust channels are opened by shifting of the cylinder in direction of the axis z (Fig. 6). The suction channels are opening after expansion of the exhaust gases (Fig. 7), together with opening of the patented cooling channel using a short delay (Fig. 8), what causes pushing of the rest exhaust gases out from the cylinder inner space.

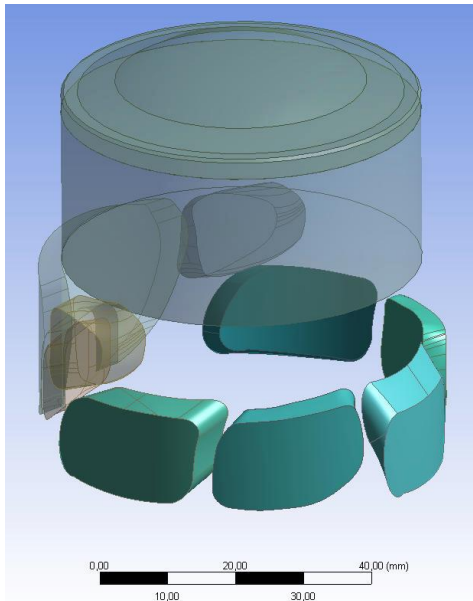


Figure 7: Suction channels.

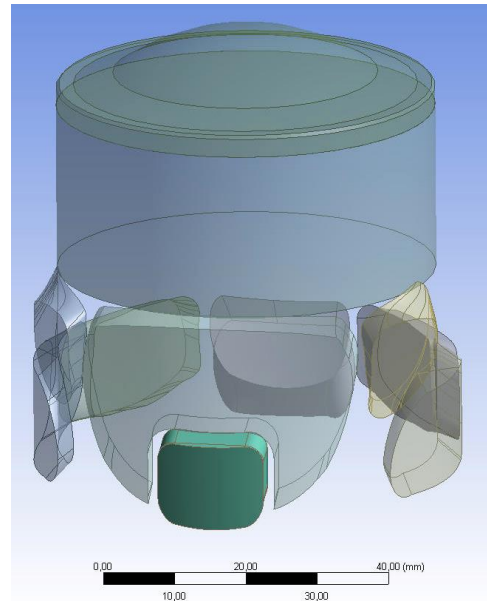


Figure 8: Patented cooling channel.

Discretisation of the domains (Fig. 9) were performed by means of three various types of the networks. The cylinder was additionally divided into the upper and bottom part with the network Sweep, which is based on the hexagonal elements [13]. The network of type sweep is suitable in the case of network deformation in order to avoid creation of negative volumes for the individual elements during the simulation computational process. The upper part of the cylinder volume uses the hexagonal network for a continual connection to the network sweep. The input and output channels are networked by means of the tetragonal network with regard to a requirement to create the boundary layers [14].

The created network consists of $1.38 \cdot 10^6$ elements and $1.27 \cdot 10^6$ nodal points, whereby a considerable amount of the elements is used just for networking of the cylinder, which is the key domain of the whole simulation.

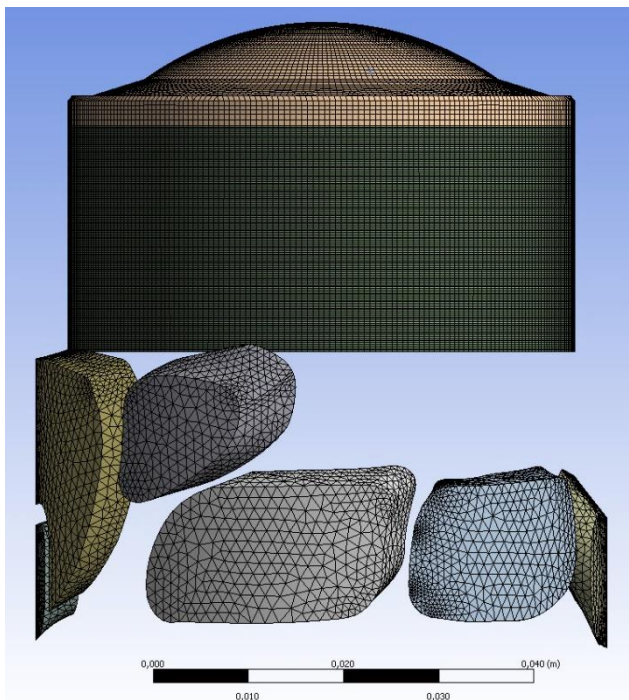


Figure 9: Discretized calculation model.

3. SETTING OF UNICITY CONDITIONS FOR SOLUTION OF SIMULATION

The geometrical conditions of unicity were defined by scanning and modification of the created model. Solution of all the applied differential equations from the area of streaming and heat transfer requires determination of the material characteristics for the domains, their initialisation conditions and boundary conditions, interfaces among the individual domains as well as conditions for the solver itself.

The material characteristics of the exhaust gases and fuel-air mixtures were determined according to stoichiometry of combustion, whereby the individual physical parameters were set as functionally depending on the temperature and pressure. In the initial time ($\tau = 0$ s), the domain of cylinder contained 100 % amount of the exhaust gases at the pressure $5 \cdot 10^5$ Pa and temperature 530 °C. The initial velocity of the gas flow in direction of the piston movement was set at the level 30.92 m/s (for $\tau = 0$ s) using the following relation, which describes the piston velocity v_p in any time τ , in order to speed-up convergence of the whole computational process:

$$v_p = \frac{dL}{d\tau} = \omega \cdot \frac{dL}{d\theta} \quad (\text{m/s}) \quad (5)$$

$$v_p = \omega \cdot a \cdot \left[\sin\left(\frac{2 \cdot \pi \cdot n}{60} \cdot \tau + \theta_4\right) - \frac{R_c}{2} \cdot \sin\left(\frac{4 \cdot \pi \cdot n}{60} \cdot \tau + 2 \cdot \theta_4\right) \right] \quad (\text{m/s}) \quad (6)$$

There was also calculated the total energy in all the domains with regard to a necessity to solve changes of the state values due to doing of a volumetric work, in accordance with the first law of thermodynamics. This requirement drastically reduced the calculation time step to the value 1 μ s in order to keep stability of convergence for the linear solver. The total solution time $2.679 \cdot 10^{-3}$ s was divided to 2679 time steps using average number of four iterations per one time-step, what means approx. 10700 iterations at relatively dense discretization of the domains. The turbulent gas flow required application of the SST (Shear Stress Transport) model, which is suitable in the case of supersonic streaming with large pressure gradients. This situation is typical mainly for the exhaust process. The possibility of network deformation was applied for the domain of cylinder. The time dependence of movement, related to the bottom surface of the cylinder inner volume, in the contact place with the piston, was defined according to Eq. (6). The cylindrical surface required defining of an „unspecified“ shift, because this surface is changing during movement of the piston.

The exhaust channel domains contained the boundary conditions for the exhaust gas outputs with a possibility of supersonic streaming. In the case of the suction channels it was applied the boundary condition for intake of the fuel-air mixture with the overpressure value 60 kPa. This overpressure is caused due to compression of mixture under the piston in the space of the piston-rod movement. It is assumed that the initial volume of mixture is compressed to 5/8, what causes 8/5-times increase of the mixture pressure, i.e. from 100 kPa to 160 kPa. The analysed time dependence of the pressure at input to the suction channels (SC) is pictured in Fig. 10.

A simplified assumption of the pressure time dependence at input to the suction channels can be modified using a more complex simulation of streaming, considering the bottom volume under the piston and the total volume of the suction channels. This more complex simulation takes into consideration dynamics of the fuel-air mixture streaming.

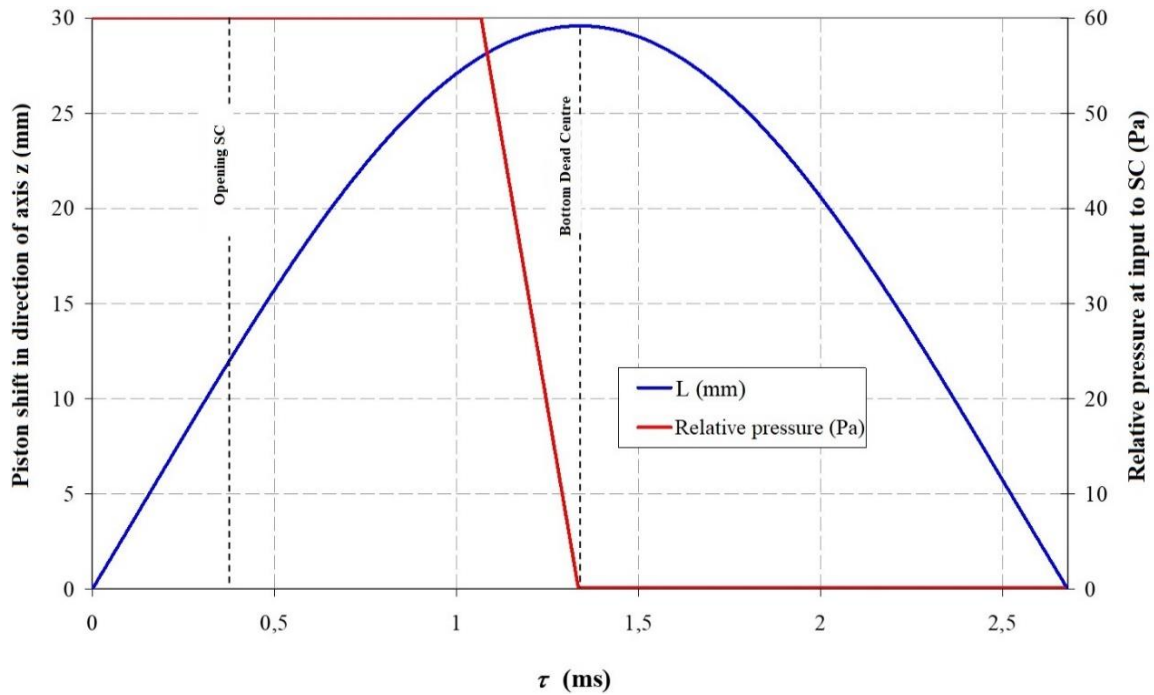


Figure 10: Assumed time dependence of relative pressure at input to SC.

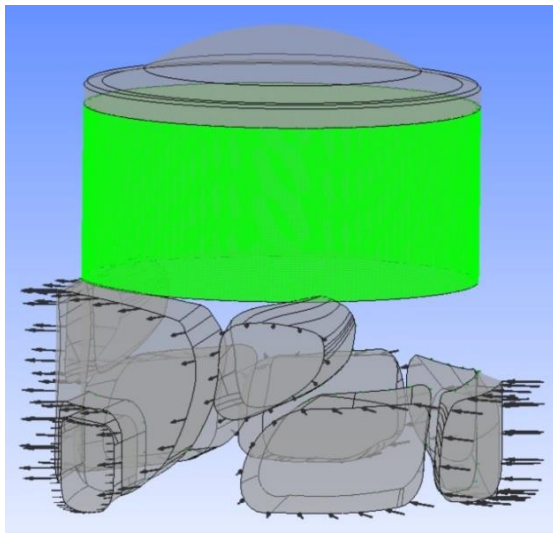


Figure 11: Illustration of interface area on cylinder.

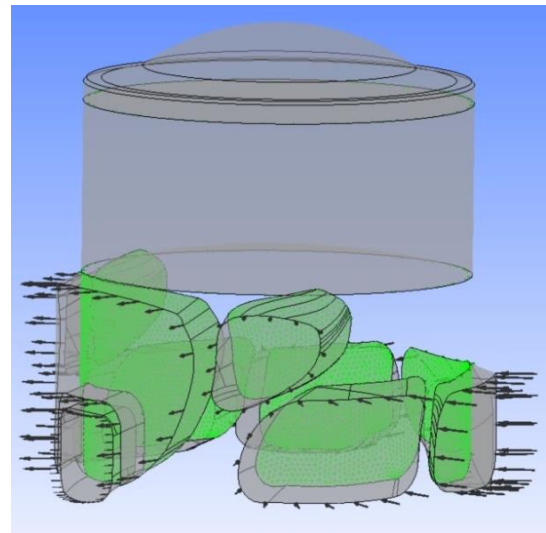


Figure 12: Illustration of interface area on EC and SC.

Since the individual domains are physically separated, it is necessary to create a domain interface among them. The domain interface is represented by the cylindrical surface of inner volume in the cylinder (Fig. 11) and by the inner cylindrical surfaces of the exhaust channels (EC) and suction channels (SC), (Fig. 12). The cylindrical surface of inner volume in the cylinder changes its size due to shift of the piston and it is in contact with the inner cylindrical surfaces of the exhaust and suction channels. The main requirement is to set a possibility for exchange of mass among the individual domains through their interface in order to enable gas flow among the domains, according to the physical principles of streaming.

It is necessary to determine the maximal number of iterations and also the iteration accuracy before starting the process of numerical solution. In spite of the predetermined maximal number of iterations 10, the predefined iteration accuracy was reached already after 4 iterations due to a short time step [15].

4. EVALUATION OF RESULTS OBTAINED FROM NUMERICAL SOLUTION USING SOFTWARE ANSYS CFX

A gradual solution of the difference equations using the iteration procedure, which is performed for the individual time steps, begins by starting of the solver. After finishing of this process it is possible to display the isosurfaces in any time step and for any position on the model. In this way can be supplemented the missing numerical values of the state values between the points 4 and 1 in the Otto cycle that cannot be obtained analytically. Fig. 13 presents the isobaric surfaces for the time moment $3.2 \cdot 10^{-4}$ s in the longitudinal cross-section of cylindrical surface. There are visible on this figure the following domains: cylinder, suction channels (bottom right), added suction channel (bottom left) and exhaust channel (top left).

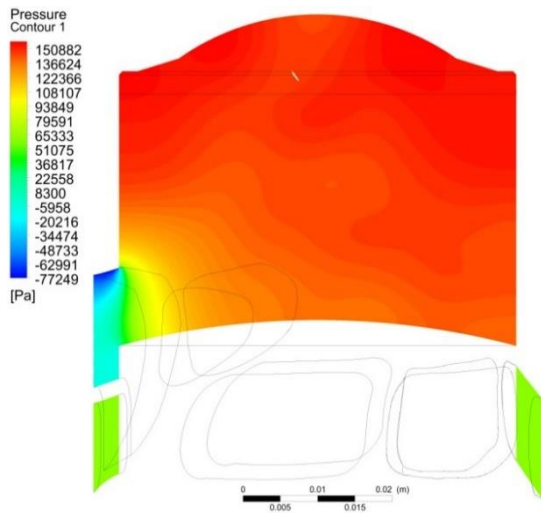


Figure 13: Isobaric surfaces in cylinder cross-section.

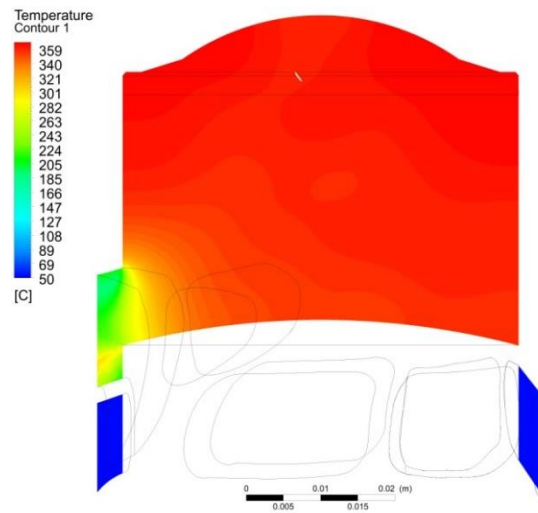


Figure 14: Isothermal surfaces in cylinder cross-section.

It is evident (from this figure) that already after turning of the crankshaft about the angle 21.13° , which corresponds to the time $3.2 \cdot 10^{-4}$ s, the pressure decreases about approx. 350 kPa due to a leakage of gas through EC and also due to increased volume of cylinder. Adequately to this pressure decrease is reduced the temperature, as well. The field of temperature in the cross-section of cylinder is visible in Fig. 14.

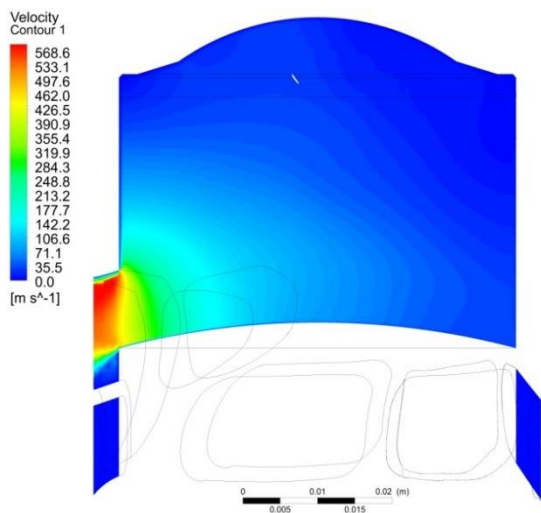


Figure 15: Field of velocities in cylinder cross-section.

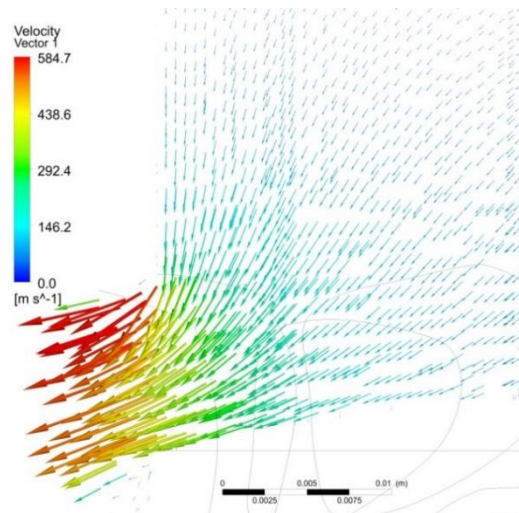


Figure 16: Detailed view on velocity vectors at place of EC in the time $3.2 \cdot 10^{-4}$ s.

Evaluation of the streaming velocities enables to display the field of absolute velocities (Fig. 15) or vectors of the velocity field (Fig. 16).

There are presented in Fig. 17 the velocity vectors during exchange of the exhaust gases by the fuel mixture in the time 0.7 ms ($\theta = 137.8^\circ$). Penetration of the fuel mixture to the exhaust gases in the place of the added SC is clearly visible.

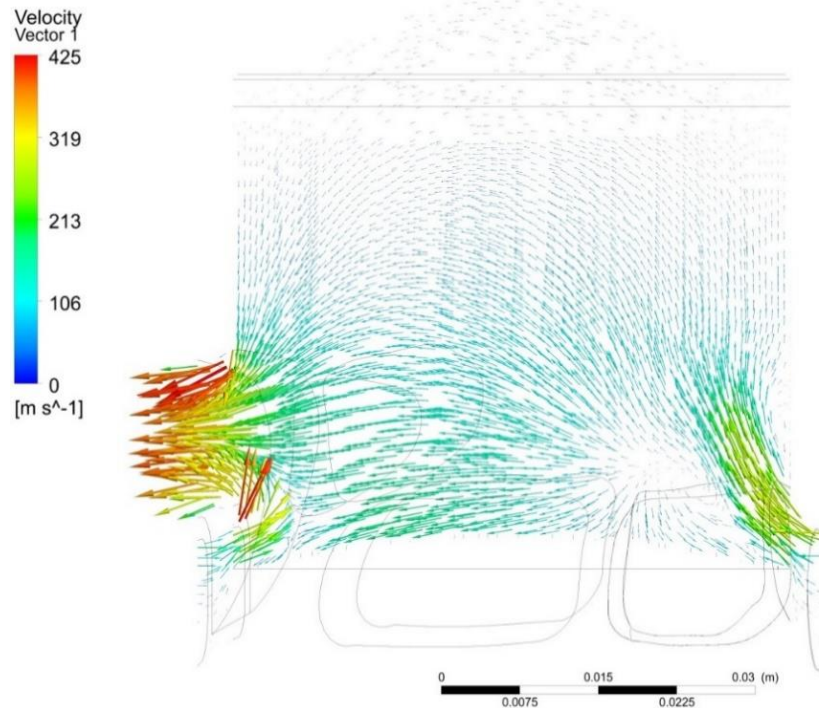


Figure 17: Velocity vectors in the time 0.7 ms ($\theta = 137.8^\circ$).

The missing part of the real Otto cycle between the points 4 and 1 can be obtained using analysis of the pressure values during the piston movement, Fig. 18.

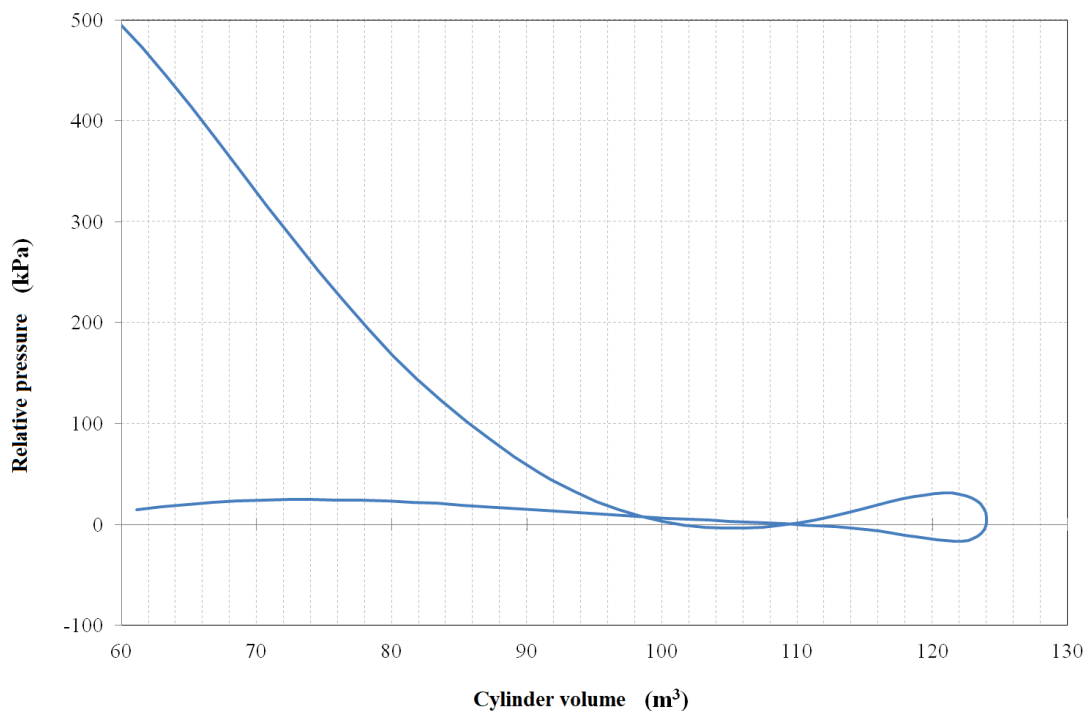


Figure 18: Diagram P-V between the points 4 and 1 in Otto cycle.

The initial phase of the exhaust channel opening causes a rapid reduction of the pressure. After the suction channel opening, the pressure in the cylinder is moderately increased, because there is an overpressure of the fresh mixture in the suction channel. Decrease of this pressure begins near the bottom dead centre. The repeated movement of the piston upwards causes a moderate increase of the pressure and in this way the rest of the exhaust gases is pushed out from EC. The dependence of temperature in the cylinder on the volume of cylinder is visible in Fig. 19.

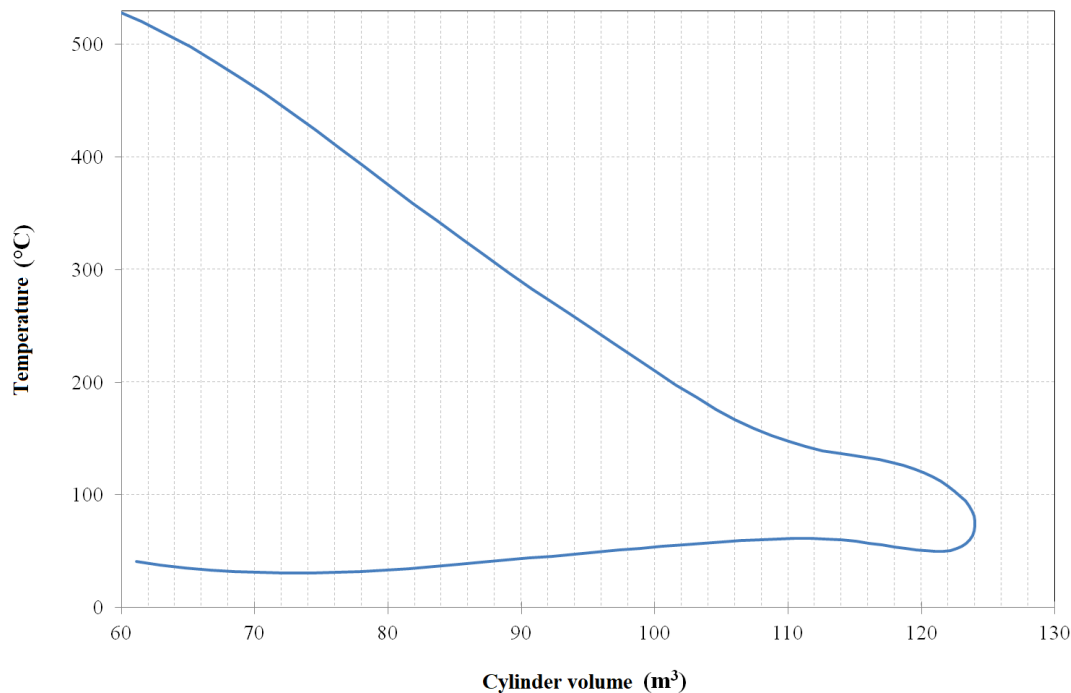


Figure 19: Dependence of average temperature in cylinder on cylinder volume.

Evaluation of the computational results enables to present a global view on the time dependence concerning discharge and suction of gases in the tested two-stroke engine, whereby it is possible to read and indirectly to determine the residual amount of the exhaust gases remaining in the cylinder as well as volume of the fresh mixture, which is blown out through EC without combustion.

5. CONCLUSIONS

The original scientific contribution of this work consists in a newly-developed system of the cooling channel, which can be applied in all the high-powerful sport applications. The system of cooling channel, which is installed in the exhaust pipe in order to eliminate the operational risks caused by application of the biofuel E85, enables to solve two important disadvantages of the high powerful combustion engines, i.e. the insufficient filling of the engine cylinder with the fresh fuel-air mixture and the excessive thermal expansivity of piston. The main advantage of this innovative technical solution consists in a fact that improved filling of the cylinder with the fresh mixture thanks to the added cooling channel increases the maximal power output of the engine. The second important advantage is that the fresh mixture, which flows to the cylinder through the cooling channel, decreases the piston surface temperature and in such a way it minimises risk of a possible destruction due to uncontrolled thermal dilatation of piston. This positive effect enables to apply also the alternative fuel mixtures, namely the biofuels, without operational problems. The presented cooling channel system was long-time developed,

really designed, successfully applied in the racing motorcycle and, consequently, it has become the subject of a patent application.

The future research plans are focused on further research and development of innovations determined for a more efficient utilisation of the renewable energy sources with regard to the planned emission limits. Our attention will be focused mainly on increasing of the efficiency during combustion of the fuels containing bio-components.

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