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The Effect of DI Engine Intake Parameters on the Gas Exchange Process

Key words: Marine four stroke engines, modelling, analysis

When hot exhaust gas is mixed with inlet air, the charge to a diesel engine is modified in different ways: the charge temperature increases, the total charge mass is reduced and the charge composition changes. This paper is concerned with the residual gas fraction that is usually determined by straight CO_2 measurement in a sample of gas extracted from a cylinder during the compression stroke. A general zero-dimensional model for calculating the residual gas fraction in reciprocating internal combustion engines has been formulated. The model accounts for both the trapped gas in the cylinder at top dead center and the back-flow of exhaust gas into the cylinder during the valve overlap period. The effect of EGR introduction has been discussed.

Efekty zmian parametrów w układzie dolotowym silnika o wtrysku bezpośrednim na proces wymiany ładunku

Słowa kluczowe: silniki okrętowe – czterosuwowe, modelowanie, analiza

Gdy gorące gazy spalinowe są zmieszane z czystym powietrzem dolotowym, ładunek ma zmienione parametry w różny sposób: wzrasta jego temperatura, masa całkowita ulega zmniejszeniu i zmienia się skład. Omówiono możliwości oceny ilości pozostałych spalin, którą normalnie oznacza się dzięki pomiarowi stężenia CO_2 w próbce gazu pobranej z komory spalania. Sformułowano podstawy teoretyczne zero-wymiarowego modelu dla obliczania ilości gazów spalinowych w silnikach tłokowych. Obliczenia prowadzi się dla fazy przepłukania w celu oszacowania ilości gazów pozostałych w komorze, jak również cofających się z układu wylotowego. Przeanalizowano możliwości realizacji recyrkulacji spalin.

Introduction

The purpose of the scavenging process is to remove the burned gases at the end of the power stroke and admit fresh charge for the next cycle. The engine gas exchange process is characterized by overall parameters such as volumetric, scavenging and trapping efficiency. These overall parameters depend on the design of the engine subsystems such as manifolds, valves and ports, as well as engine operating conditions. During the induction process, pressure losses occur as the inlet air or exhaust gas passes through or by each of these components. There is an additional pressure drop across the intake and exhaust valve. The marine engine exhaust system typically consists of an exhaust manifold, turbocharger, exhaust pipe and a muffler or silencer. The gases flows are pulsating. However, many aspects of these flows can be analyzed on a quasi-steady basis. The drop in pressure along the intake and exhaust system depends on engine speed, the flow resistance of the elements in the system, the cross-sectional area through which fresh charge and exhaust gases move and their density. When engine is turbocharged, due to the time-varying valve open area and cylinder volume, gas inertia effects, and wave propagation in the intake and exhaust systems, the pressures in the intake, the cylinder and the exhaust during the gas exchange process vary in a complicated way. Analytical calculation of these processes is difficult. In practice, these processes are often treated empirically using overall parameters such as volumetric efficiency to define intake and exhaust system performance.

Three types of models for calculating details of intake and exhaust flows have been developed and used:

1. Quasi-steady models for flow through the restrictions – valve and ports
2. Filling and emptying models, which account for the finite volume of critical manifold components
3. Gas dynamic models which describe the spatial variations in flow and pressure throughout the manifold

Each of these types of models can be useful for analyzing engine behavior. The appropriate choice depends on objectives, and the time and effort available. The second aspect has been investigated by this author previously and reported in references [1].

1. Intake and exhaust flows models – gas dynamic

Many inductions and exhaust system design variables determine overall performance. These flow paths are often simplified by considering the flow to be one-dimensional. In general, the gas is highly turbulent and fluid frictional forces are present within the fluid and the walls. In simple analyses, the fluid is assumed to be in viscid and wall frictions are allowed for by use of a friction factor.

Further, the gas is taken to be perfect (with constant specific heat) for the examination of many flow processes. These variables include the length and cross-sectional area of both primary and secondary runner, the entrance and exit angles. Most of this geometric detail is beyond the level which can be incorporated into the model.

Coupled with the pulsating nature of the flow into and out of cylinder, these details create significant gas dynamic effects on intake and exhaust flows which require a more complex modelling approach. Gas dynamic models have been in use to study engine gas exchange process. These models use the mass, momentum and energy conservation equations for the unsteady compressible flow in the intake and exhaust. Normally, the one-dimensional unsteady flow equation is used. Also, often models use a thermodynamic analysis of the in-cylinder processes to link the intake and exhaust flows. The method of characteristics is used to solve the gas dynamic equations [2], [3]. Finite difference techniques are used in more recent intake and exhaust flow models.

1.1. Unsteady flow equations

Consider the flow through the control volume within a straight duct shown in Fig. 1. It is assumed that the area change over the length *dx* of the control volume is small so the flow is essentially one-dimensional. The continuity, momentum and energy equations are developed for the control volume.

An equivalent diameter for the duct is given by:

$$D = \sqrt{\frac{4F}{\pi}} \tag{1}$$

So that:

$$\frac{1}{F}\frac{dF}{dx} = \frac{2}{D}\frac{dD}{dx} \tag{2}$$

The continuity equation states: the net rate of flow out the control volume equals the rate of decrease in mass in the control volume. The rate of mass flow entering the control surface is: $\rho u F$.

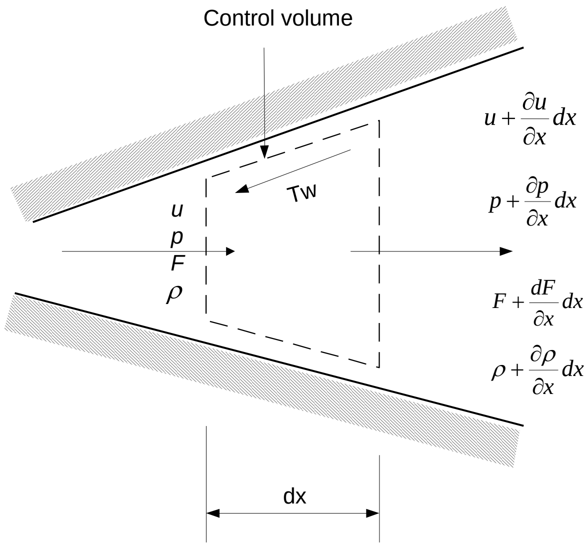

Fig. 1. The control volume for unsteady one-dimensional flow analysis
Rys. 1. Elementarna objętość dla zero-wymiarowej niestacjonarnej analizy przepływu

The rate of mass flow leaving the control surface is:

$$\left(\rho+\frac{\partial p}{\partial x}dx\right)\left(u+\frac{\partial u}{\partial x}dx\right)\left(F+\frac{dF}{dx}dx\right) \tag{3}$$

The rate of decrease of mass within the control surface is:

$$-\frac{\partial}{\partial t}(\rho F dx) \tag{4}$$

Hence, substituting equitation 4 into the continuity equation gives:

$$\left(\rho+\frac{\partial p}{\partial x}dx\right)\left(u+\frac{\partial u}{\partial x}dx\right)\left(F+\frac{dF}{dx}dx\right)-\rho uF=-\frac{\partial}{\partial t}(\rho F dx) \tag{5}$$

This simplifies to:

$$\frac{\partial(\rho uF)}{\partial x}dx=-\frac{\partial}{\partial t}(\rho F dx) \tag{6}$$

To first-order small quantities, or expanding and rearranging:

$$\frac{\partial p}{\partial t} + \rho \frac{\partial u}{\partial x} + u \frac{\partial \rho}{\partial x} + \frac{\rho u}{F}\frac{dF}{dx} = 0 \tag{7}$$

The momentum equation states: the pressure forces and shear forces on the control surface equal the rate of momentum within the control volume and the net efflux of momentum out of control surface. The pressure forces on the control surface are equal to the sum of the forces on the end faces and the force on the side walls – all forces are assumed positive. The shear forces are due to friction at the wall. The forces are then given by the following expressions – pressure and shear forces:

$$pF - \left(p + \frac{\partial p}{\partial x}dx\right)\left(F + \frac{dF}{dx}dx\right) + p\frac{dF}{dx}dx = -\frac{\partial}{\partial x}(pF)dx + p\frac{dF}{dx}dx \tag{7}$$

$$-\tau_w \pi D dx = -f\frac{\rho u^2}{2}\pi D dx \tag{8}$$

where:

D – equivalent diameter,
τ_w – wall shear stress,
f – friction factor.

Hence, the momentum equation gives:

$$u\left(\frac{\partial p}{\partial t} + \rho\frac{\partial u}{\partial x} + \frac{\partial u}{F}\frac{dF}{dx} + u\frac{\partial p}{\partial x}\right) + f\frac{\rho u^2}{2}\frac{4}{D} + \frac{\partial p}{\partial x} + \rho\frac{\partial u}{\partial t} + \rho u\frac{\partial u}{\partial x} = 0 \tag{8}$$

Based on first law of thermodynamics for control volume the energy equation becomes:

$$q\rho F dx = \frac{\partial}{\partial t}\left[(\rho F dx)\left(C_v T + \frac{u^2}{2}\right)\right] + \frac{\partial}{\partial x}\left[(\rho u F)\left(C_v T + \frac{u^2}{2} + \frac{p}{\rho}\right)\right]dx \tag{9}$$

The entropy change suffered by the particle as it passes through the control volume along the path line can be obtained from the second law in the form:

$$Tds = d(C_v T) + pd\left(\frac{1}{\rho}\right) \tag{10}$$

These expressions (7), (8), (9) and (10) are the conservation equations for one-dimensional non-steady flow.

One of the techniques used to solve these equations is characteristics method with a numerical accuracy that is first order in space and time, and requires a large number of computational points if resolution of short-wavelength variations is important.

1.2. Method of characteristics

This method is well established mathematical technique for solving hyperbolic partial differential equations. The partial differential equations are transformed into ordinary differential equations that apply along the so-called characteristics lines. Pressure waves are the physical phenomenon of practical interest in the unsteady intake flow, and these propagate relative to flowing gas at the local sound speed. In this application, the one-dimensional unsteady flow equations (listed above), are rearranged so that they contain only local fluid velocity and sound speed. Thus, the solution of the mass and momentum conservations for this one-dimensional unsteady flow is reduced to the solution of set ordinary differential equations. The equations are solved numerically using rectangular grid. The intake or exhaust system is divided into individual pipe sections which are connected at junctions. A mesh is assigned to each section of pipe between sections. Gas pressure, density, and temperature can then be calculated from energy conservation equation and the ideal gas law.

2. Engine used in prediction and experimental details

The scheduled program of basic measurements was carried out on a test-bed medium speed engine (Table 1) operating at steady conditions for speed and load. For each load and speed setting, the engine performance data were recorded. Some essential operating data were measured in accordance with ISO-3046 standard. Amongst other variables, these included: effective power, speed, fuel consumption, exhaust gas temperature, condition of the turbo blowers, together with the ambient conditions prevailing at the time of the measurement. The mode of operation determined using the engine test cycles relevant for generators propulsion, as specified in ISO standards 8178 part 4. The engine was supplied by means of distillate fuel – ISO-F-DMA.

As the valve and port together is usually the most important flow restrictions and alternatively, valve events can be defined based on angular criteria along the lift curve. What is important is when significant gas flow through the valve-open area either starts or ceases. The instantaneous valve flow area depends on valve lift and the geometric details of the valve head, seat, and stem. Intake and exhaust valve open areas corresponding to a typical valve-lift profile are plotted versus crankshaft angle in Fig. 2.

Table 1

Test engine specification
Dane silnika eksperymentalnego

Engine		Nominal rate	
Designation	Type	Power [kW]	Speed [revs/min]
Generator	Sulzer – 6A20D	540	900

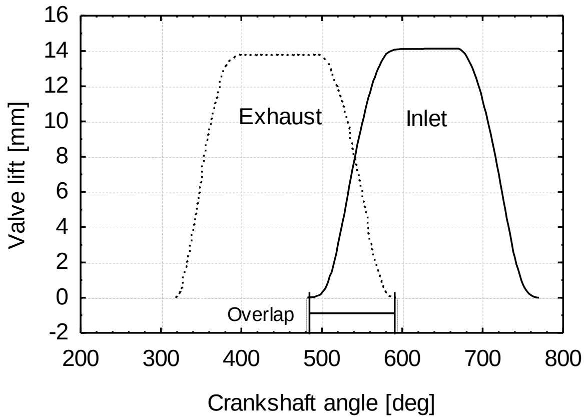

Fig. 2. Valve timing diagram for Sulzer A20D engine
Rys. 2. Wykres wartości wzniosów zaworowych dla silnika Sulzer A20D

Prediction was carried out using an experimental engine test results presented in Table 2. The exhaust gas mass flow and combustion air consumption are based on exhaust gas concentration and fuel consumption measurement. Universal method, known as carbon/oxygen-balance, which is applicable for fuels containing H, C, S, O, N in known composition is used. Exhaust gas composition depends on the relative proportions of fuel and air fed to the engine, fuel composition, and completeness of combustion. The overall combustion reaction can be written as:

$$\begin{gathered} \mathrm{HC_x} + A\left(\mathrm{O_2} + 3.7274\mathrm{N_2} + 0.0444\mathrm{Ar} + 0.0014\mathrm{CO_2}\right) + B\mathrm{H_2O} \Rightarrow \\ \Rightarrow a\mathrm{CO} + b\mathrm{CO_2} + c\mathrm{O_2} + d\mathrm{H_2O} + e\mathrm{HC_x} + f\mathrm{H_2} + g\mathrm{N_2} + h\mathrm{NO} + j\mathrm{Ar} \end{gathered} \tag{11}$$

where:

$B = 4.7733\ A\ (P_s/P_a - P_s)$,

P_s – saturation vapour pressure of the inlet air [N/m^2],

P_a – ambient barometric pressure [N/m^2].

Then the balance equations are as follows:

carbon – C:

$$1 = a + b + e \tag{12}$$

hydrogen – H:

$$x + 2B = 2d + xe + 2f \tag{13}$$

oxygen – O:

$$2A + B = 2b + 2c + d + h \tag{14}$$

Table 2

Measured data and calculated gas emission of a Sulzer A20D engine

Wyniki pomiarów i obliczeń emisji spalin dla silnika Sulzer A20D

Test cycle /Mode	D2	1	2	3	4	5
Power	%	100	75	50	25	10
Speed	%	100	100	100	100	100
SFOC	g/kWh	225.6	222.3	229.9	281.2	439.0
Fuel flow	kg/h	127.18	94.16	65.40	41.39	26.99
Air flow	kg/h	4 106.0	3 191.0	2 253.0	1 588.0	1 261.0
Exhaust flow	kg/h	4 234.0	3 285.0	2 318.0	1 629.0	1 288.0
NO_x mass flow	kg/h	5.72	4.80	3.33	1.88	1.10
CO mass flow	kg/h	0.29	0.23	0.20	0.17	0.16
CO_2 mass flow	kg/h	402.6	298.0	206.9	130.9	85.3
O_2 mass flow	kg/h	507.4	409.7	292.9	222.6	196.7
THC mass flow	kg/h	0.0022	0.0018	0.0012	0.0009	0.00068
SO_2 mass flow	kg/h	0.18	0.13	0.09	0.06	0.04
NO_x specific	g/kWh	10.15	11.32	11.70	12.80	17.87
T_a	kPa	100.5	100.5	100.5	100.5	100.5
t_{inlet}	°C	27.0	26.0	25.0	23.0	22.0
H_a	%	78.0	76.0	75.0	74.0	73.0

An investigation was conducted with the aim of identifying and quantifying the effects of charge air temperature and gas recirculation (EGR) on residual fraction in cylinder trapped charge. The effects of EGR were investigated: the reduction in oxygen supply to the engine, participation in the combustion process of carbon dioxide and water vapour present in the EGR, increase in the specific heat capacity of the engine inlet charge, increased inlet charge temperature and reduction in the inlet charge mass flow rate arising from the use of hot EGR.

3. Results and discussion

The present work was aimed at quantifying the effects on cylinder charge residuals of, firstly, increasing the inlet charge temperature and, secondly, decreasing charge mass. In diesel engines, when exhaust gas is mixed with inlet air the charge trapped in the engine cylinders at the start of the compression stroke is modified in a number of ways [4]:

1) the charge temperature can be considerably higher than the charge temperature when only air is being admitted to the cylinders.
2) the mass of the trapped charge is reduced owing to the higher temperature of the charge and the associated reduction in charge density.
3) The charge composition is substantially different from the composition of air.

Figure 3 shows that the residual mass fraction in trapped charge mass declines substantially when engine load increases from idle to 50 per cent, and inclines up to the maximum under gradually increasing load.

The gravimetric residual mass left over from the previous cycle strictly depends on engine load that is shown in Figure 4.

The increase in charge temperature can affect combustion in a number of ways. For example, the peak combustion temperature can be significantly higher, which increases NO_x production and exhaust emissions; the ignition delay can also be reduced, with consequent effects on several exhaust pollutant emissions. Figures 5 and 6 show the changes in residual gas ratio brought by the progressive rise of inlet charge air temperature, from 280 K to 330 K. It can be seen that the residual mass fraction fell to a minimum level, when the charge air temperature was only 280 K and under partial engine load.

Figure 6 shows that the residual mass rise approximately linearly when the engine load is increased, while keeping the residual mass approximately steady constant with inlet charge temperature change. This rise in cylinder gas temperatures was also reflected in higher exhaust gas temperatures. In general, Fig. 5 and 6 show that the residual mass amount was not increased considerably by heating of the inlet charge.

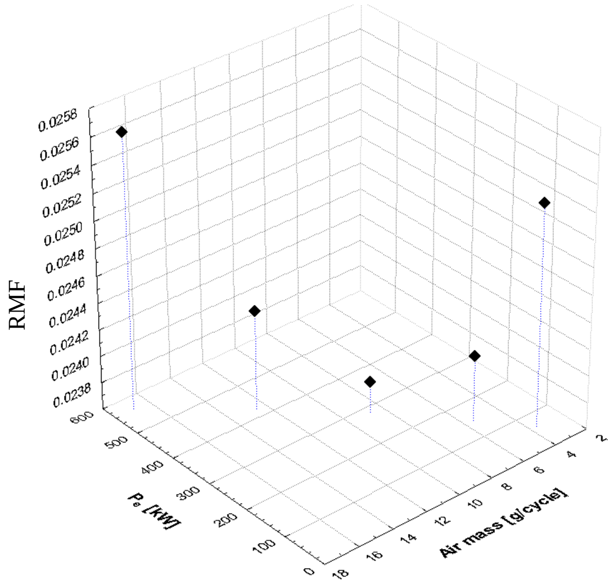

Fig. 3. Influence of engine load on residual mass fraction in trapped air
Rys. 3. Wpływ obciążenia silnika na ilość spalin resztkowych w ładunku

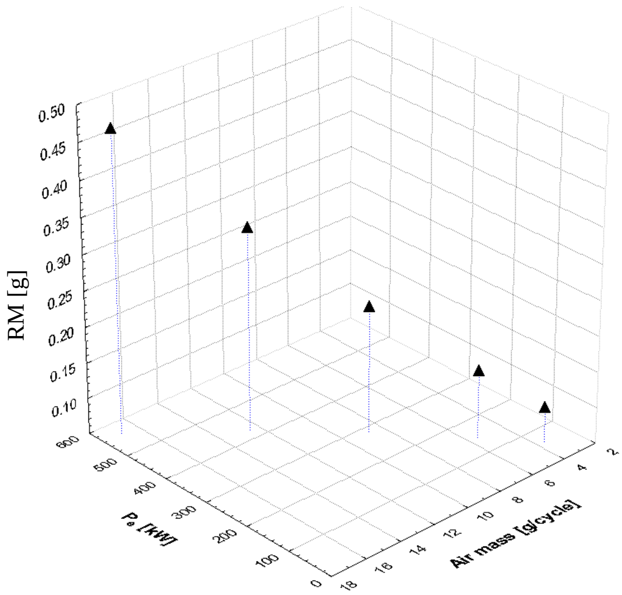

Fig. 4. Influence of engine load on gravimetric residual mass in trapped air
Rys. 4. Wpływ obciążenia silnika na ilość masy spalin resztkowych w ładunku

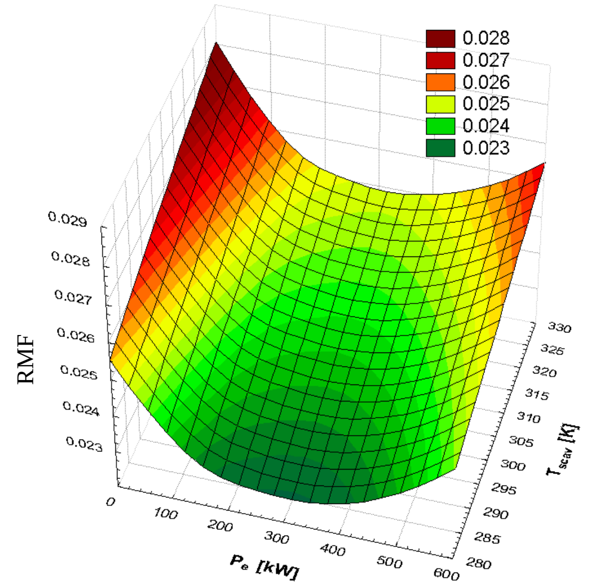

Fig. 5. Influence of engine load and charge air temperature on RMF in trapped air
Rys. 5. Wpływ obciążenia silnika i temperatury powietrza na pozostałość spalin

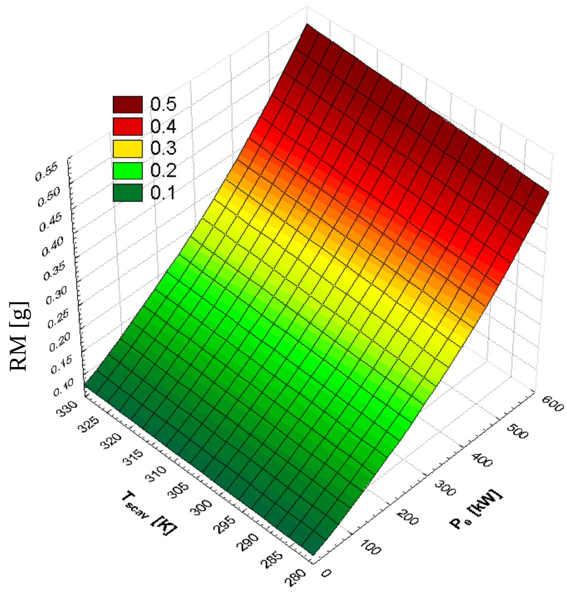

Fig. 6. Influence of engine load and charge air temperature on RM in trapped air
Rys. 6. Wpływ obciążenia silnika i temperatury powietrza na masę pozostałości spalin

As mentioned above, the mixing of exhaust gas with inlet air alters substantially the composition of the trapped charge by comparison with air. In some engines a fraction of the engine exhaust gases is recycled to the intake to dilute the fresh mixture for control of NO_x emission. This is because exhaust gas recirculation has a substantial concentration of carbon dioxide and water vapour, in addition to nitrogen and oxygen. As a result, the trapped charge contains a lower concentration of oxygen than air, a roughly similar concentration of nitrogen and some carbon dioxide and water vapour. The reduction in the amount of oxygen has a major impact on combustion and emissions. For example, it increases substantially the amount of combustion-generated soot escaping oxidation, but can also lower NO_x generation [5, 6].

If the percentage of exhaust gas recycled (% EGR) is defined as the percentage of the total intake mixture which is recycled exhaust:

$$EGR(\%) = \left(\frac{m_{EGR}}{m_{inlet}} \right) \cdot 100 \tag{15}$$

where:

m_{EGR} – mass of exhaust gas recycled.

Then, the residual mass gas fraction, in fresh mixture is:

$$RMF = \frac{m_{EGR} + m_r}{m_{cyl}} \tag{16}$$

Figure 7 illustrates the way in which residual mass increased rapidly with increasing EGR rate and engine load. The figure was drawn using results calculated for the engine conditions shown in Table 2. Figure 8 shows that the residual mass fraction in trapped charge mass declines when the EGR fraction increases from 0 to 15 per cent (by mass). The reduction in the trapped charge mass can also affect combustion in a number of ways. For example, the heat absorbing capacity of the charge is reduced on account of its lower mass and this can result in higher combustion temperatures. Also, the reduction in charge mass lowers the amount of oxygen available for combustion of the fuel; this is because the amount of oxygen trapped in the cylinders is reduced.

Clearly, at this engine running condition, the use of EGR should be limited to around 25 per cent (by mass), since higher levels would reduce the oxygen - fuel ratio below that available in diesel engines at full load operation without EGR. EGR levels higher than 25 per cent (by mass) could be employed at lighter engine loads, when more oxygen is generally available in the engine cylinder.

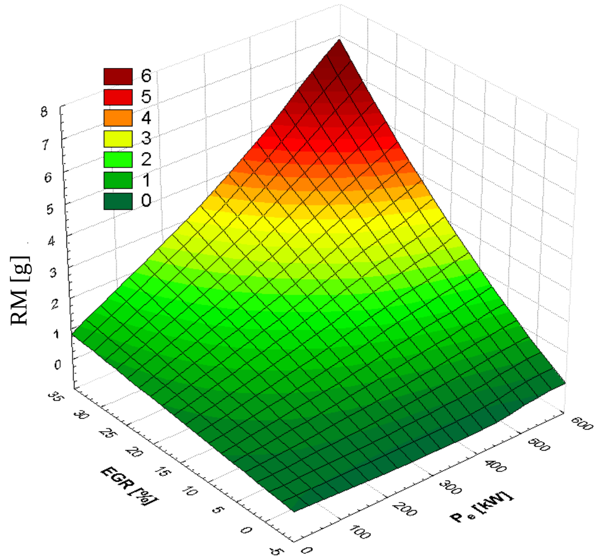

Fig. 7. Influence of engine load and EGR on RM in trapped air
Rys. *7. Wpływ obciążenia silnika i EGR na masę pozostałości spalin*

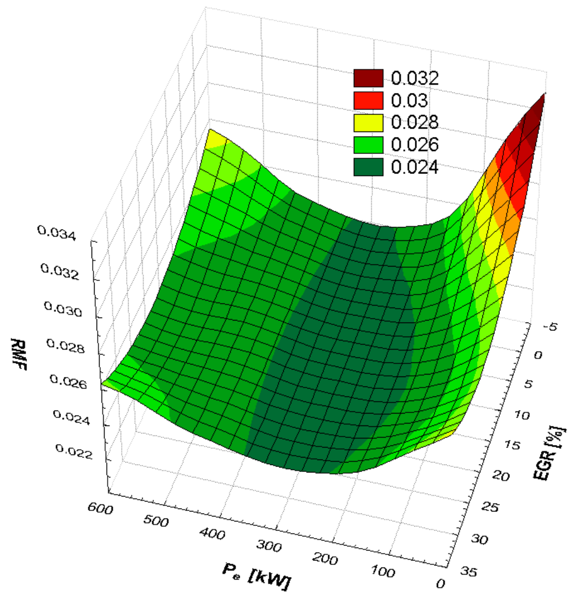

Fig. 8. Influence of engine load and EGR on RMF in trapped air
Rys. 8. Wpływ obciążenia silnika i EGR na pozostałość spalin

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Symbols

h – specific enthalpy,
m – mass,
$\dot{m}$ – mass flow,
p – pressure
t – time,
u – velocity,
φ – fuel-air equivalence ratio,
ρ – density,
C_v – specific heat at constant vol.
D – diameter,
EGR – exhaust gas recirculation,
F – area of cross-section,
R – gas constant,
RM – residual mass,
RMF – residual mass fraction,
T – temperature,
V – volume,

Subscripts

a – ambient,
cyl – cylinder,
egr – exhaust gas recirculation,
$inlet$ – inlet,
r – residual.

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