BULETINUL INSTITUTULUI POLITEHNIC DIN IAŞI
Publicat de
Universitatea Tehnică „Gheorghe Asachi" din Iaşi
Tomul LXI (LXV), Fasc. 1, 2015
Secţia
CONSTRUCŢII DE MAŞINI

PHYSICO-MATHEMATICAL MODELING OF THE INTAKE PROCESS IN INTERNAL COMBUSTION ENGINE (I)

BY

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Received: January 28, 2015
Accepted for publication: February 20, 2015

Abstract. Intake system of an internal combustion engine is the connection between the engine cylinder and the external air source, being the system that provides qualitative and quantitative adjustment of the proportion of the air or the air-fuel mixture required by the engine cycle processes. From the dynamic point of view, the intake system of an ICE is the component in which is found a spectrum of pressure variation conditions, that determine the evolution of characteristic parameters such as maximum torque or maximum effective power for a given motor shaft rotation regime.

The most important role of the intake system is to provide the maximum amount of mass of air or fuel mixture required to obtain determined values of the parameters (maximum torque and maximum power output, in close connection with the speed of the engine).

Key words: intake process; inlet; intake manifold.

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1. Introduction

The paper presents a different approach to the ways of calculating the fuel intake system components of internal combustion engines. In order to study the chosen physical model it needs to analyze the indicated diagram of the internal combustion engine, corresponding to the intake process.

This diagram shows the variations of characteristic parameters that influence the intake process. It was considered, as the initial hypothesis, that the main driving force of the intake process is the cylinder piston movement between the two dead points, TDC and BDC.

The piston movement is made with a variable speed, from zero to the maximum value, determined by the movement of the crank radius with a certain angle of rotation of the crankshaft α_{RAC}. By increasing the volume generated by the piston movement between TDC and BDC, the pressure in the cylinder decreases. This causes a pressure difference between the ends of the load in fresh intake manifold, which will determine its movement in negative pressure.

To model intake process easier for an internal combustion engine, we made a number of simplifying assumptions. Thus, it is considered as follows:

− the intake process is isotherm;

− fresh load flow along the route of intake is one-dimensional;

− the working fluid is incompressible;

− external pressure cylinder is considered the atmospheric pressure;

− the temperature at which the process takes place is equal to the air temperature;

− fresh load flow along the intake route occurs under the effects of the depression created by the piston movement between the two dead points, **TDC**, **BDC**;

− the pressure in the cylinder at the beginning of the intake process is equal to atmospheric pressure;

− the working fluid is considered as a perfect stoichiometry mixture of air and gasoline;

− fresh load flow takes place without losses;

− fresh load flow takes place from the outside to the inside of the cylinder.

2. Physico-Mathematical Modeling of the Intake Process

To define the physical and mathematical model of the intake process we need to define the engine constructive parameters. There are required also a number of adjustment parameters of components that are part of the motor, which intervene and influence the evolution of the intake process.

Thus, we define the following structural features: **D** – cylinder bore; **S** – piston stroke; **l** – the length of the connecting rod ; **r** – crank radius; $\mathbf{V_t}$ – total engine capacity; 9 $\mathbf{V_s}$ – unitary engine capacity; $\mathbf{V_a}$ – combustion chamber volume; **ε** – volume ratio; **λ = r/l** – the ratio between the length of the connecting rod and the crank radius; D_{sa} – inlet valve diameter; d_{0a} – diameter of the hole of inlet valve; $\mathbf{d_{ga}}$ – diameter of intake manifold; α_{adsa} – angle of advance to the opening of the intake valve; α_{iisa} – angle of delay when closing the inlet valve; h_{sa} – height of the inlet camshaft lift; n – engine speed; α_{RAC} – crankshaft rotation angle; $\Delta\alpha_a$ – rotation angle during the intake process.

The duration of the intake process is given in eq. (1):

$$\tau_a = \frac{\Delta\alpha_a}{6 \cdot n} \tag{1}$$

The distance of movement for the piston from **TDC** according to crankshaft rotation angle is given in eq. (2):

$$x = r\left[\left(1 - \cos\alpha_{RAC}\right) + \frac{\Lambda}{4}\left(1 - \cos 2\alpha_{RAC}\right)\right] \tag{2}$$

If the diameter of the cylinder and piston stroke are selected, then in a first step of modeling are determined the geometric size of the valve plate, the free orifice size and lift height of the valve as follows:

$$D_{sa} = 0.44 \div 0.55\ D \tag{3}$$

$$d_{oa} = 0.88 \div 0.93\ D_{sa} \tag{4}$$

$$h_{samax} = 0.25 \div 0.3\ D_{sa} \tag{5}$$

Since the flow rate of the mixture through the inlet valve area is very important, its maximum value influencing the flow at certain speeds, it is necessary to check the proper selection of the diameter value by using the relationship of continuity of flow to the cylinder and the valve level.

Starting from the equality equation of flows at the entrance to the piston section for a given speed range, we can write

$$S_p \cdot w_p = S_{sa} \cdot w_{samax}\ , \tag{6}$$

which allows an approximation of the S_{sa} (theoretical value for the flow section of the valve, eq. (7)):

$$S_{sa} = \frac{S_p \cdot w_p}{w_{samax}} \tag{7}$$

Knowing that
$$S_p = \frac{\pi \cdot D^2}{4}, \ [m^2] \tag{8}$$
$$S_{sa} = \frac{\pi \cdot D_{sa}{}^2}{4}, \ [m^2] \tag{9}$$
$$w_p = \frac{S \cdot n}{30}, \ [m/s] \tag{10}$$

it can be determined the maximum speed of the mixture through the gate w_{samax} valve as in eq. (11):

$$w_{samax} = \frac{\frac{\pi \cdot D^2}{4} \frac{S \cdot n}{30}}{S_{sa}} = \frac{\pi \cdot D^2 \cdot S \cdot n}{120 \cdot S_{sa}} = \frac{\pi \cdot D^2 \cdot S \cdot n}{120 \frac{\pi \cdot D_{sa}^2}{4}} = \frac{D^2 \cdot S \cdot n}{30 \cdot D_{sa}^2}, \quad [m/s] \tag{11}$$

In practice, there are not admitted maximum speeds mixture whose values exceed 110 m/s; from the previous relationship, for a given engine with geometric dimensions and at a high speed, the speed may exceed the maximum permissible value. Therefore, to reduce the maximum values of the w_{samax} we need to choose a section S_{sa} to reduce the speed limit. That makes the filling carried out in optimal conditions at high speed regimes but it would have a totally inappropriate development at low revs, due to the reduced value of inlet velocity along the route and in the inlet gate valve.

In order to physico-mathematical model of the filling process, we had to choose a generic geometric configuration of the inlet path to allow particular modeling of all possible technical solutions (Gaiginschi, 1995; Kumar, 1999; Ismail & Bakar, 2008; Befrui, 1994).

For these reasons we designed schematization as in Fig. 1.

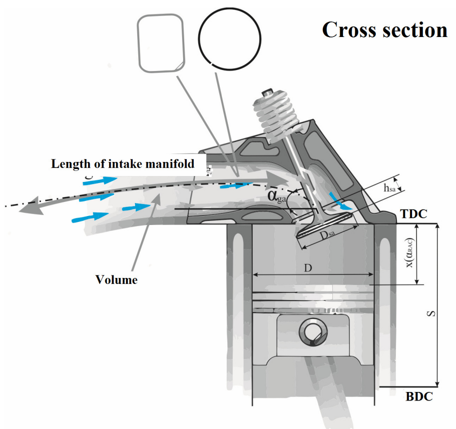

Fig. 1 − Schematization of the mathematical model.

It can be seen that the geometrical dimensions of the inlet path defines a volume of fluid. With the diameter of the hole, a characteristic of the valve size above, and with transverse configuration of inlet manifold designed to ensure minimum gaso-dynamic loss, we determine the length of the inlet. This geometric dimension is designed in the assumptions of inertional filling and the resonance of intake route considered as a sound tube. To be able to determine the resonance frequencies of the route of intake, it was necessary first to choose the speeds.

As a rule, for designing of the inlet manifold should, be selected as the reference speed the maximum torque speed. As current trends in the design of internal combustion engines has evolved, you can choose two or even more speeds for sizing the inlet galleries. This is possible due to the development of technical solutions to change the length of the manifold according to the engine revolution. For these reasons, the physical and mathematical modeling of the intake process is made for two revs: the engine maximum torque value speed, and respectively speed for maximum power.

The intake path can be assimilated as a sound tube, which evolves in two distinct situations. A situation is characteristic for the time interval in which the inlet valve is closed, in which case the intake is equated with a sound tube with one end closed and another situation, feature starts filling process, in which case the intake is equated with a sound tube with both ends open, case assimilated and functioning as a Helmholtz resonator.

3. Physico Mathematical Model Development

With the lengths of the intake path chosen and knowing the diameter of the gallery can determine the volume occupied by the fresh load. Thus, we can express this mathematically as follows (eqs. (12) and (13)):

$$V_{Mmax} = L_{Mmax} \cdot S_{ga}, [m^3] \tag{12}$$

$$V_{Pmax} = L_{Pmax} \cdot S_{ga}, [m^3] \tag{13}$$

Taking into account the density of the work fluid we obtain a fresh load mass characteristic for the two chosen speed regimes. With this size, the quantity of fresh load, we will determine the filling process characteristic equations.

In this way we obtain a physical system that can be represented as in Fig. 2.

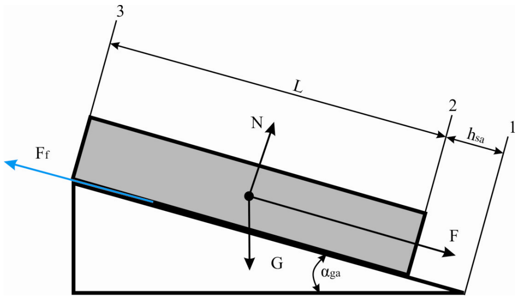

Fig. 2 − Physical model of the intake manifold.

In order to study the physical model chosen it needs to analyse the indicated diagram of the internal combustion engine, which is reserved for the intake process.

In this diagram you can see the variations of characteristic quantities that influence the process of filling. It was considered, as the initial hypothesis, the main driving force of the process of filling - the cylinder piston movement, between the two dead points TDC and BDC.

The piston movement is made with a variable, speed from zero to the maximum value, determined by the movement of the crank radius with a certain angle of rotation of the crankshaft α_R. By increasing the volume generated by the piston movement between those two points, the pressure in the cylinder decreases. This causes a pressure difference between the ends of the load in fresh intake manifold, which will determine its movement in negative pressure.

To be able to analyze these details in the Fig. 3 is plotted the engine indicated diagram feature intake process, in conjunction with the piston speed variation between the two points, and variation in the speed of fresh cargo flow through the valve cylinder. We've marked with $\mathbf{w_a}$ – speed of flow of the mixture in the cylinder, $\mathbf{w_p}$ – piston speed, $\mathbf{\hat{i}_a}$ – the beginning of filling, 1 – engine block; 2 – cylinder; 3 – connecting rod; 4 – crankshaft; 5 – exhaust manifold; 6 – inlet manifold; 7 – exhaust valve; 8 – inlet valve; 9 – camshaft; 10 – the piston; 11 – the spark plug; p_0 – atmospheric pressure; p_a – inlet pressure.

From the analysis on the diagram of the variation of the velocity of flow of fresh load in the cylinder, there is a delay of working fluid flow while plunger reaches a certain speed and volume of the cylinder rose from $\mathbf{V_a}$ a $\mathbf{V_{s\,i}}$. This delay is due to the force of inertia and the difference in pressure between the two ends of enclosed volume for fresh load in the gallery and the intake valve port.

To set in motion this mass of fresh fluid must overcome its inertia force. Analyzing the physical model proposed above, we can write the equations that define the minimum force required. We consider the mass of fluid that is located on an inclined plane since, as a rule, intake manifolds have a certain inclination to the horizontal α_{ga} between 0 ÷ 60°. Thus, on the mass **m** of fresh fluid acts the force of its own weight **G** and a force of friction $\mathbf{F_f}$. The mass value can be obtained with eq. (14). For the mass **m** to be at rest, eqs. (15) and (16) are written.

$$m = \rho \cdot L_{ga} \cdot S_{ga}, \ [kg] \tag{14}$$

$$F = 0 \tag{15}$$

$$F = G - F_f; \quad F = m \cdot g \cdot \sin\alpha_{ga} - \mu \cdot m \cdot g \cdot \cos\alpha_{ga} \tag{16}$$

To overcome the force of inertia of mass **m** must apply, from one end of it, a force at least equal to the force **F**.

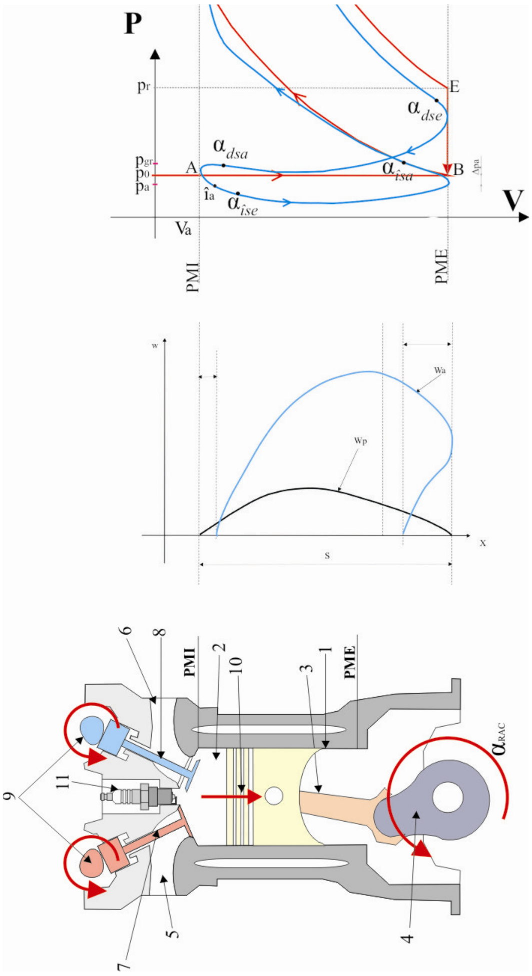

Fig. 3 − Engine indicated diagram feature intake process.

As noted earlier, the force that sets in motion the fresh load is the result of the difference in pressure between the cylinder and surface gallery offered the flow valve. Knowing that, mathematically, the pressure is equal to the ratio of the force acting perpendicular over a specific surface: $p = \frac{F}{S}$. We can consider that the difference in pressure between the outside and the cylinder, reported to the minimum area of the inlet valve, can generate the necessary force to overcome the force of inertia of the fresh load (eq. (17))

$$F = \Delta p_{min} \cdot S_{min_{sa}} \tag{17}$$

where
$$\Delta p_{min} = p_0 - p_{cil} \tag{18}$$

The minimum flow section of the inlet valve is a variable that depends on the angle of rotation of the crankshaft α_{RAC} and lifting height of the valve h_{sa}. Thus, the minimum area of the intake valve, for values of $\mathbf{h_{sa} \leq 0.123\ D_{to}}$ is given in eq. (19):

$$S_{min} = \pi \cdot h_{sa} \left(D_{sa} - 2 \cdot p + \frac{h_{sa}}{2} \cdot \sin\alpha \right) \cdot \cos\alpha \tag{19}$$

where **p** is the width of the face of the valve seat and **α** the angle of its settlement**.** Thus, the value of the minimum necessary force to overcome the inertia force of the fresh load becomes (eq. (20)):

$$F_{min} = F = \Delta p_{min} \cdot \pi \cdot h_{sa} \left(D_{sa} - 2 \cdot p + \frac{h_{sa}}{2} \cdot \sin\alpha \right) \cdot \cos\alpha = m \cdot g \cdot \left(\sin\alpha_{ga} - \mu \cdot \cos\alpha_{ga} \right) \tag{20}$$

Equaling the value of the two forces we will get eq. (21):

$$F_{min} = F = \Delta p_{min} \cdot \pi \cdot h_{sa} \left(D_{sa} - 2 \cdot p + \frac{h_{sa}}{2} \cdot \sin\alpha \right) \cdot \cos\alpha = m \cdot g \cdot \left(\sin\alpha_{ga} - \mu \cdot \cos\alpha_{ga} \right) \tag{21}$$

In this eq. (21) we have several unknowns, mainly dependent on the angle of rotation of the crankshaft. The momentary height of lift of the intake valve is dependent on the movement of the piston and the angle of rotation of the crankshaft. Also the difference of pressure between the cylinder and the valve flow offered is dependent on the angular movement of the crankshaft.

To find out at what the value of depression occurs overcoming the inertia force of fresh load, must know the momentary lifting height of the valve, which generates the minimum section of the fluid flow valve.

If we introduce the simplifying assumption that the intake process shall start from the moment when the pressure in the cylinder is equal with pressure in valve inlet port, which takes place after a period of angled crankshaft, and fresh fluid flow will only be made in the direction of port valve-cylinder, we can approximate the minimum lifting height of the valve in order to meet the necessary movement of fresh load.

As we know, the opening of the inlet valve is made with an angular position upfront **TDC.** This advance is necessary due to the inertia of the mechanism of distribution but also due to the need of taking the heat of gap cam and valve or valve and their bearings. Whereas the valve opening takes place before the process of forced discharge is completed, the pressure in the cylinder is superior to that of the inlet. There is the possibility that a certain amount of flue gas to penetrate into the inlet. However, it is possible that due to pressure waves generated to the closing of the intake valve, the pressure in the manifold would be superior to that of the cylinder so that the reverse flow cannot occur. In this hypothesis we mathematically modeled the filling process.

In practice, the hypothesis is viable since intake process may not take place until the time of completion of the condition that $p_{cil} \leq p_{ga}$. This is possible only after the piston has reached the position of **TDC**, the crankshaft has come by default value equal to the angular advance to the opening of the intake valve $\alpha_{a\,dsa}$. To find out the value of the momentary lifting height of the inlet valve h_{sa}, corresponding to **TDC** position, we can choose two versions, one that takes into account the knowledge of the lift valve law depending on crankshaft rotation and design of valve profile, and another based on planimetric mapping.

Because the method of determination of h_{sa} on the basis of the law of waiver is simple, we will address the solution to determine the height of the lifting of the valve through the planimetry. So, if you know the law of variation of the section of the momentary valve offered depending on the maximum height of the valve lift and angle range of crankshaft in the intake process, and knowing that the maximum value of the lifting of the valve is obtained at approximately half $\Delta\alpha_a$ using the graphic we can determine the approximate value of h_{sa} corresponding to **TDC** position. Thus we can calculate the minimum section of fresh mixture flow valve. However, according to **TDC**'s position, fresh gas flow in cylinder does not take place, the speed of travel of the piston in the cylinder, $\mathbf{w_p}$, is zero and so the gas velocity at the gate valve, $\mathbf{w_{sa}}$. The actual flow of the mixture occurs with the beginning of movement of the piston in the cylinder.For the filling process to be carried out efficiently, it is necessary for the beginning of the process to take place near the **TDC** with the start of the race of the piston downward by **BDC**.

Whereas the momentary lifting height of the valve, at angle area $\alpha_{a\,dsa}$ must compensate for the heat gaps of the components and valve mechanism, we impose that when reaching **TDC** the momentary valve lift height has a given value, which will lead to the determination of the minimum section provided by the valve.

The law lifting height of the valve is sinusoidal type and if you impose its maximum value on the vertical axis and consider the value of angular intake process on the horizontal axis we obtain a representation as in Fig. 4.

Thus, for a chosen value of lifting height $\mathbf{h_{sai}}$ we have the camshaft valve minimum section S_{min} at an angular position $\Delta\alpha_{ai}$ as in eq. (22).

$$S_{min\,i} = \pi \cdot h_{sa\,i}\left(D_{sa} - 2\cdot p + \frac{h_{sa\,i}}{2}\cdot \sin\alpha\right)\cdot \cos\alpha\,, \quad [m^2] \qquad (22)$$

Bernoulli equation, writtten for an incompressible fluid between section 1 and 2 of the physically system examined, *i.e.* between the cylinder and the valve inlet port will get the value that will move fresh load (eq. (23)):

$$p_{ga} + 0 = p_{psa} + \frac{\rho}{2}\cdot w_{psa}^2\,, \quad [m/s] \qquad (23)$$

With the help of these relations we can determine the minimum required value for overcoming the inertia of fresh gas. Momentary speed of the piston $w_{\alpha_{RAC}p}$ can be determined based on the eq. (24):

$$w_{\alpha_{RAC}p} = \frac{\pi\cdot n\cdot r}{30}\left(\sin\alpha_{RAC} + \frac{\Lambda}{2}\sin 2\alpha_{RAC}\right)\,, \quad [m/s] \qquad (24)$$

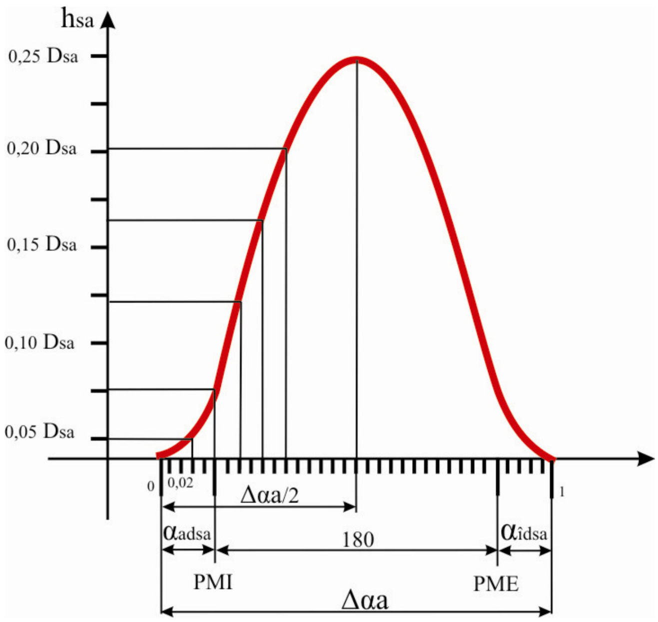

Fig. 4 − Lifting height depending on the crank angle by planimetry.

Writing the equation of conservation of energy, applied between the section provided by opening the valve and piston head we obtain (eq. (25)):

$$w_{sa} \cdot S_{min} = w_p \cdot S_p;\; w_{sa} \cdot S_{min} = \frac{\pi \cdot n \cdot r \cdot S_p}{30}\left(\sin\alpha_{RAC} + \frac{\Lambda}{2}\sin 2\alpha_{RAC}\right), \text{[m/s]} \quad (25)$$

To get a small value of delay of the beginning of intake, we choose an elementary rotation interval of the crankshaft $\Delta\alpha_{ai}$ with small values of the order of 0.01 $\Delta\alpha_a$.

Considering the angle of advance to the opening of the intake valve as necessary for taking the heat gap, the lifting height of the valve will be adequate to achieve uniformity in the TDC by the piston. To approximate the value of the height of lifting, at the beginning of the filling process, we divide the range of angular rotation of the crankshaft, which is reserved for lifting and closing the valve at a number of $9 \cdot \frac{h_{sa}}{D_{sa}}$. We get a integer ψ incremental value of 0.01 in $\Delta\alpha_{RAC\,i}$, as follows (eq. (26)):

$$\alpha_{RAC\,i} = 0.01 \cdot \psi \cdot \Delta\alpha_{RAC\,i} \quad (26)$$

Knowing that the maximum amount of intake valve lift takes place at mid-race piston, we raise a normal line on the horizontal axis in the middle of the range, determining the angular position for the maximum lifting height. It splits the lifting height at smaller units that comply with the step $\frac{h_{sa}}{D_{sa}}$. We place these on the vertical axis and then a horizontal line intersects the sinusoidal curve that shapes the valve lift law. From the intersection points obtained we plot vertical lines that will intersect the horizontal axis with values of angular rotations of the crankshaft. The corresponding values are determined by the appropriate lifting angle α_{RAC}.

Choosing the lowest lift height shall be calculated on the minimum section of the valve for the following condition (eq. (27)):

$$\frac{h_{sa}}{D_{sa}} \leq 0.123; \qquad S_{min} = \pi \cdot h_{sa}\left(D_{sa} - 2 \cdot p + \frac{h}{2} \cdot \sin\alpha\right) \cdot \cos\alpha \quad (27)$$

Substituting in eq. (21), we obtain the minimum amount Δp_{min} required to move the fresh load (eq. (28)):

$$\Delta p_{min} = \frac{m \cdot g \cdot \left(\sin\alpha_{ga} - \mu \cdot \cos\alpha_{ga}\right)}{\pi \cdot h_{sa\,i}\left(D_{sa} - 2 \cdot p + \frac{h_{sa\,i}}{2} \cdot \sin\alpha\right) \cdot \cos\alpha} \tag{28}$$

The time elapsed from the beginning of opening the inlet valve until the beginning of filling is equal to:

$$\tau_{\hat{i}a} = \frac{\alpha_{adsa} + \alpha_{RAC\,i}}{6 \cdot n}, \quad [s] \tag{29}$$

and the remaining time of the intake process is:

$$\tau_{ra} = \frac{\Delta\alpha_a - \alpha_{adsa} - \alpha_{RAC\,i}}{6 \cdot n}, \quad [s] \tag{30}$$

To be able to determine the pressure in the valve inlet port and then the pressure in the cylinder, corresponding to changes in the values of incremental angle of rotation of the crankshaft, you need to define the boundary conditions that occur in sections 3-2, the free end of the gallery and port valve, *i.e.* sections 2-1, the port and cylinder valve. This section represents a restriction in the path column of fresh load thus generating flow conditions at speeds higher than the speed of sound.

Due to the fact that the flow velocity of mixture of fresh fluid can reach, under certain conditions, the speed of sound, which lead to severe debt limitation of fluid circulating through the valve orifice, it is defined the critical pressure ratio the pressures to be equal to (eq. (31)):

$$\frac{p_{psa}}{p_{ga}} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{31}$$

For values of the adiabatic coefficient k = 1.4, corresponding to air, we obtain a value of the critical pressure ratio $\frac{p_{psa}}{p_{ga}} = 0.528$ and for the case of an air-gas mixture in the stoichiometry report unit, k = 1.353 and $\frac{p_{psa}}{p_{ga}} = 0.536$.

To provide detailed explanations we have Fig. 5. Since we made the assumption that the flow is taking place from intake manifold in the cylinder and not vice versa, it appears that the pressure in the cylinder $\mathbf{p_{cil}}$ is greater than the critical pressure, flow regime is subcritic, otherwise imposed by limiting the speed of flowing fresh cargo through the port of the valve to 110 m/s.

Considering the flow of fresh load as similar to a perfect gas and fluid flow, and considering intake process as a isentropic process, can write flow characteristic equations for temperature and pressure as follows (eq. (32) and (33)):

$$T_0 = T + \frac{w^2}{2 \cdot c_p} \tag{32}$$

$$\left(\frac{T}{T_0}\right) = \left(\frac{p}{p_0}\right)^{\frac{k-1}{k}} \tag{33}$$

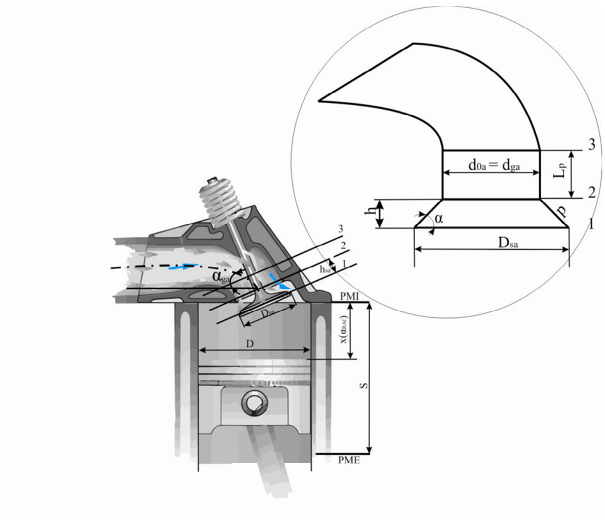

Fig. 5 − Valve-cylinder limit area.

Comparing the flow velocities of working fluid with the speed of sound **c**, by that environment, we can determine **Mach** number, M, as the ratio between the actual flow rate of the fluid and the speed of sound in that medium (eq. (34)).

$$M = \frac{w}{c} \tag{34}$$

where
$$c = \sqrt{k \cdot R \cdot T_0} \tag{35}$$

We can write the eqs. (36) and (37) as follows:

$$\frac{T}{T_0} = 1 + \frac{k-1}{2} \cdot M^2 \quad (36)$$

$$\frac{p_0}{p} = \left(1 + \frac{k-1}{2} \cdot M^2\right)^{\frac{k}{k-1}} \quad (37)$$

where **p** respectively **T** are absolute temperature and pressure parameters corresponding to the ordered section valve, **psa** noted in this study.

The value of pressure in the section controlled by the valve can be represented as in eq. (38):

$$p_{psa} = p_{ga}\left(1 + \frac{k-1}{2} \cdot M^2\right)^{-\frac{k}{k-1}} \quad (38)$$

Mach number value can be easily determined at the time of the angular movement of the crankshaft, causing momentary value of the flow rate of the valve port by using the equation of conservation of energy in the **1**-cylinder valve, respectively.

$$W_{psa} \cdot S_{min} = w_p \cdot S_p ; \quad W_{\alpha_{RAC} p} = \frac{\pi \cdot n \cdot r}{30}\left(\sin\alpha_{RAC} + \frac{\Lambda}{2}\sin 2\alpha_{RAC}\right) \quad (39)$$

$$w_{psa} = \frac{\pi \cdot n \cdot r \cdot S_p}{30 \cdot S_{min}}\left(\sin\alpha_{RAC} + \frac{\Lambda}{2}\sin 2\alpha_{RAC}\right) \quad (40)$$

To determine the value of the fresh load pressure in the port valve (corresponding to the position 2 in Fig. 5), we write Bernoulli equation, for an incompressible fluid, flowing with friction through a pipeline from a pressure equal to atmospheric pressure to pressure $\mathbf{p_{psa}}$ (eqs. (41)-(43)).

$$p_0 = p_{ga} + \frac{\rho}{2} \cdot w_{ga}^2 \quad (41)$$

$$w_{psa\,i} \cdot S_{min\,i} = w_{ga} \cdot S_{ga} ; \quad S_{ga} = \frac{\pi \cdot d_{ga}^2}{4}, \ [m^2] \quad (42)$$

$$w_{ga} = w_{psa} \frac{4S_{min}}{\pi \cdot d_{ga}^2}, \ [m/s] \quad (43)$$

$p_{ga} = p_0 - \frac{\rho}{2} w_{ga}^2$ the equation that defines the pressure value in inlet manifold depending on the speed of movement of fresh load through pipeline.

Once initiated the intake process, change of characteristics can be made by using the equation of conservation of energy and the general equation of perfect gases, written for the intake manifold areas-port valve into the cylinder, as follows in eqs. (44)-(46).

$$w_{psa\,i} \cdot S_{min\,i} = w_{ga} \cdot S_{ga} \tag{44}$$

$$w_{psa\,i} \cdot S_{min\,i} = Q_{psa\,i} \tag{45}$$

is the volumetric flow rate at the moment of the intake valve;

$$w_{ga} \cdot S_{ga} = Q_{ga} \tag{46}$$

is currently volumetric flow generated by the movement of the piston in the cylinder. The pressure acting on the valve port area is the pressure in the cylinder $p_{psa\,i} = p_{cil\,i}$.

By limiting the maximum amount of flow velocity through the section of the valve at a value of 110 m/s, we have at all times a subcritic flow, which can cause the fluid flow through the associated section of valve opening as follows. We can thus determine the mathematical expression of the mass flow rate of the fluid flow through the intake (eq. (47)):

$$m = \rho \cdot S_{sa} \cdot w_{psa}, \text{ [kg/s]} \tag{47}$$

Expressing the flow velocity based on Mach number, depending on the density of the pressure $\mathbf{p_0}$ and $\mathbf{T_0}$, adiabatic coefficient, atmospheric pressure and absolute temperature of ambient, we obtain the theoretical flow perpetuated by the section of the valve with the following eq. (48):

$$m_{sa} = S_{min\,sa} \frac{p_0}{\sqrt{RT_0}} \left(\frac{p_{psa}}{p_0}\right)^{\frac{1}{k}} \sqrt{\frac{2k}{k\text{-}1}\left[1-\left(\frac{p_{psa}}{p_0}\right)\right]^{\frac{k-1}{k}}}, \quad \text{[kg/s]} \tag{48}$$

With the help of these mathematical relationships we can analyze the evolution of flow velocities in the inlet and the gallery, in the section provided for inlet valve flow, depending on the value of the height of the valve lift and the angular movement of the crankshaft.

To get a physical-mathematical modeling, in greater detail, in terms of the evolution of fresh load in the cylinder, using the indicated diagram and the variation curves of the velocity, of motion of the piston speed and fresh load, according to the angular movement of the crankshaft, we obtain the characteristic equations of evolution speeds along the intake process.

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MODELAREA FIZICO-MATEMATICĂ A PROCESULUI DE ADMISIE AL UNUI MOTOR CU ARDERE INTERNĂ (I)

(Rezumat)

Sistemul de admisie al unui motor cu ardere internă reprezintă legătura dintre cilindrul motorului şi sursa de aer exterior, sistem care asigură reglarea cantitativă şi calitativă a proporţiei de aer sau amestec aer-combustibil necesară desfăşurării proceselor ciclului motor.

Din punct de vedere dinamic, sistemul de admisie al unui motor cu ardere internă, este elementul constructiv în care se regăseşte un spectru de condiţii ale variaţiei de presiune, care determină evoluţia unor parametri caracteristici cum ar fi momentul motor maxim sau puterea efectivă maximă, pentru un anumit regim de rotaţie a arborelui motor.

Rolul cel mai important al sistemului de admisie este acela de asigurare a unei cantitaţi maximale de masă de aer sau de amestec carburant necesar obţinerii unor valori determinate ale parametrilor, moment motor maxim şi putere efectivă maximă în strânsă legătură cu regimul de turaţie al motorului.