INFLUENCE OF THE ROTOR SHAPE ON THE FLOW RATE TRANSPORTED BY A MACHINE WITH PROFILED ROTORS

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Abstract. The paper presents a new type of rotating machine with profiled rotors; each rotor has two rotating pistons that can have the following shapes: I – rectangular blade; II – isosceles triangle; III – curvilinear (oval). The shape of the piston influences the modification of the flow rate transported by each rotor; the theoretical flow rate transported by the machine is calculated for the I^st case; calculation relations that prove the decrease of the flow rate in the II^nd and III^rd cases are subsequently deduced.

Keywords: rotasting machine, profiled rotors.

1. INTRODUCTION

Profound study of theoretical aspects and finding of new solutions regarding the transformation of various forms of energy in other forms of energy constitute problems that are very much on the map at present.

The paper intends to present a new patent-based [1] type of working rotating machine with profiled rotors.

This machine can be used in various ways:

- as working machine (pump, compressor, fan);
- as force machine (steam engine, hydrostatic engine).

A working machine, the low pressure compressor, was chosen for analysis. It is known from specialty literature [2] [3] [4] that the compressors classify in two categories: volumetric and dynamic ones.

Volumetric compressors refer to piston compressors and rotating compressors.

Rotating compressors are characterized by the rotating movement of one or more rotors provided with blades, rotating or special shape pistons.

Rotating volumetric compressors classify in three categories in function of the compression process nature [2] [3] [4] [5]:

a) rotating volumetric compressors with inside compression;

b) rotating volumetric compressors with outside compression;

c) rotating volumetric compressors with comound compression.

The constructive solution analyzed in the paper classifies in the b) category.

Gas compression succeeds when the volume of gas transported by the rotor from intake to discharge is connected to the discharge space where the gas from the discharge collector (with pressure p_r) flows inside the space where the low pressure (p_a) gas is found. Subsequently, the gas compressed this way is evacuated by the rotating piston towards the discharge collector. The pressure of the gas discharged by these compressors is limited by the manufacturing accuracy imposed to the moving parts (in direct contact), accuracy that establishes the size of the clearance where with the "reverse" gas flow happens, from the high pressure space (discharge) to the low pressure space (intake).

2. THE CONSTRUCTIVE SOLUTIONS

Three constructive solutions that can be used as low pressure compressor (blower) are presented. Each of the three solutions has two rotors, each rotor has two rotating pistons of different shapes. The following versions can be distinguished:

I - I^st version: Rotating pistons having the shape of a rectangular blade;

II – II^nd version: Triangular shape rotating pistons;

III – III^rd version: Curvilinear shape rotating pistons.

The sketch and the functioning principle for the three versions of the new type of rotating compressor (NTRC) are presented in continuation.

2.1. Rotating volumetric compressor with rotating pistons in the shape of rectangular blades

This case is only theoretical, namely the com-ressor is ideal, the thickness of the rectangular blade is neglectable and does not influence the flow rate transported by the compressor (fig.1).

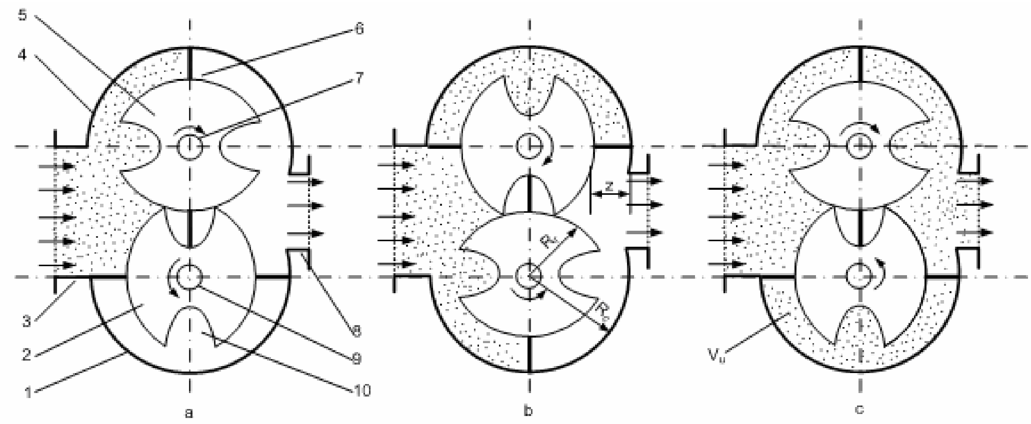

Fig.1 – Section through the Ist version of rotating machine.
1 – lower casing; *2* – lower rotor; *3* – intake chamber; *4* – upper casing; *5* – upper rotor; *6* – rotating piston in shape of rectangular blade; *7* – driven shaft; *8* – discharge chamber; *9* – driving shaft; *10* – cavity where the upper rotor piston enters.

The compressor is constituted of two sub-ssemblies:

- The mobile subassembly comprises the rotors (2) and (5) attached on the shafts (7) and (9). Two rectangular blades that constitute the rotating pistons are fixed on each rotor; the four blades separate the low pressure section (intake) of the high pressure section (discharge).

- The fixed subassembly consists of two half cylinder casings (1) and (4) endowed with intake branch pipe (3) and discharge branch pipe (8). Two useful volumes V_u are transported from intake to discharge during the rotation of each rotor. A useful volume is established by the area (A) of the section and the rotor length (l) (dimension normal to the plane of the figure):

$$\dot{V}_u = A \cdot l = \left(\frac{\pi \cdot R_c^2}{2} - \frac{\pi \cdot R_r^2}{2} \right) \cdot l \qquad (1)$$

The aspirated fluid (fig. 1.a.) is transported towards discharge and, after a 90° rotation of both rotors, positions from figure 1.b. and, subsequently, 1.c. are reached.

After a 180° rotation, the fluid that exists in the useful volume V_u (fig.1.c), namely the space encompassed by the pistons, the lower casing (1) and the lower rotor (2), will be sent to the discharge chamber. Two such volumes [2] [3] [4] [5] will be transported from intake to discharge during a complete rotation of the shaft (9):

$$\dot{V}_u = 2 \cdot \left(\frac{\pi \cdot R_c^2}{2} - \frac{\pi \cdot R_r^2}{2} \right) \cdot l \ \text{[m}^3\text{/rot]} \qquad (2)$$

The radius of the casing (R_c) equals the sum of the rotor radius (R_r) and the piston height (z):

$$R_C = R_r + z \ \text{[m]} \qquad (3)$$

$$\dot{V}_u = \pi \cdot l \cdot z \cdot (z + 2 \cdot R_r) \ \text{[m}^3\text{/rot]} \qquad (4)$$

The volumetric flow rate discharged by a sole rotor of length l [m] and rotation speed n [rot/min] will be:

$$\dot{V}_u = \pi \cdot l \cdot z \cdot (z + 2 \cdot R_r) \cdot \frac{n}{60} \ \text{[m}^3\text{/s]} \qquad (5)$$

Because the machine has two identical rotors, the flow rate transported by the machine in the Ist version will be:

$$\dot{V}_{m,I} = 2 \cdot \dot{V}_u = \pi \cdot l \cdot z (z + 2 \cdot R_r) \cdot \frac{n}{30} \ \text{[m}^3\text{/s]} \qquad (6)$$

If the following values are considered: l = = 0.05 [m]; z = 0.03[m]; R_c = 0.08 [m] and n = 300 [rot/min], it results:

$$\dot{V}_{m,I} = \pi \cdot 0.05 \cdot 0.03 \cdot (0.03 + 2 \cdot 0.05) \cdot \frac{300}{30} = 22.042 \ \text{[m}^3\text{/h]} \qquad (7)$$

It is obvious that, for the Ist version, it is difficult to attach the blade to the rotor; the case of a theoretical compressor was analyzed. Relations for computing the flow rate of the gas transported by the machine were established in the hypothesis that the blade thickness was neglectable. For the IInd and the IIIrd versions of machine, the transported flow rate will decrease because the volumes occupied by the body of the triangular, respectively curvilinear piston, will be subtracted from the useful volume V_u.

2.2. Rotating volumetric compressor with rotating pistons in the shape of triangular blades

The compressor is composed (fig.2) of two special shape identical rotors (2,5) that rotate with the same speed inside two casings (1,4); the synchronous rotation of two rotors is guaranteed by the use of a gearing consisting of two helical spur gears mounted inside or outside the machine.

The spur gears have the same pitch diameter and are mounted on the shafts 7 and 9; they assure a rotation movement such as the rotating pistons (6) of the upper rotor enter inside the cavities (10) of the lower rotor.

2.3. Rotating volumetric compressor with curvilinear rotating pistons

The only difference from the IInd version is the shape of the rotating pistons; the functioning of the machine is similar to the IInd version.

The advantages of this version - compressor with rotating pistons of curvilinear (oval) shape – refer to the delimitation between the low pressure area and the high pressure area; the delimitation is thus guaranteed by the means of the following contacts:

a) between the round tip of the pistons and the casings;

b) between the round tip of the upper rotor piston and the cavity of the lower rotor (fig.3); this contact is permanently maintained following three lines:

- between the round tip of the piston and the lower rotor;

- between the two edges of the cavity and the rotating piston of the upper rotor; the edges of the cavity are in permanent contact with the surface of the piston.

The two contacts (a) and (b) are better than in the case of the Ist and IInd versions, leading to the decrease of the flow rate by „reverse flow".

3. ESTABLISHING THE RELATIONS USED FOR THE COMPUTATION OF THE FLOW RATE TRANSPORTED BY THE MACHINE

For the rotating volumetric compressor with rotating pistons in the shape of rectangular blades, the computation relation of the transported flow rate was previously established (in section 2.1).

In the case of the rotating volumetric compressor with triangular pistons, the useful volume V_u, figure 4, will be diminished with a volume denoted $V_{p,\text{II}}$.

For a rotor, during a rotation, the volume V_u is diminished with (fig.4.):

$$V_{p,II} = 4 \cdot S_{ABC} \cdot l = 4 \cdot \frac{1}{2} \cdot \left(z \cdot \frac{b}{2} \right) \cdot l = z \cdot b \cdot l \quad \left[\text{m}^3/\text{rot} \right] \tag{8}$$

The flow rate transported by a rotor will be:

$$\dot{V}_u = \left[\pi \cdot l \cdot z \left(z + 2 \cdot R_r \right) - \dot{V}_{p,II} \right] [\text{m}^3/\text{rot}] \tag{9}$$

$$\dot{V}_u = \left[\pi \cdot l \cdot z \cdot \left(z + 2 \cdot R_r \right) - z \cdot b \cdot l \right] [\text{m}^3/\text{rot}] \tag{10}$$

The machine has two rotors:

$$\dot{V}_u = 2 \cdot \dot{V}_u \cdot \frac{n}{60} \ [\text{m}^3/\text{s}] \tag{11}$$

$$\dot{V}_{m,II} = \left[\pi \cdot l \cdot z \cdot (z + 2 \cdot R_r) - z \cdot b \cdot l \right] \cdot \frac{n}{30} \ [\text{m}^3/\text{s}] \tag{12}$$

Calculations specific to strenghth of materials proved that, for a duraluminium rotor and a discharge pressure of 3bar, the base (b) of the piston must be b = 3 cm. For the same data as in the case of the Ist version, b = 0.03 m and the same speed, the following flow rate is obtained:

$$\dot{V}_{m,II} = \left[\pi \cdot 0.05 \cdot 0.03 \cdot 0.03 + 2 \cdot 0.05 \cdot 0.03 \cdot 0.03 \cdot 0.05 \right] \cdot \frac{300}{30} = 20.422 \ [\text{m}^3/\text{h}] \tag{13}$$

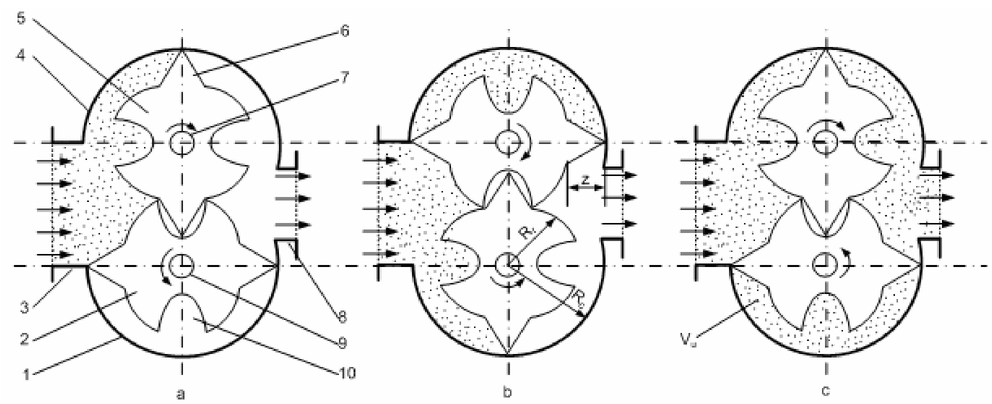

Fig. 2. The functioning principle of the IInd version of rotating volumetric machine:
1 – lower casing; *2* – lower rotor; *3* – intake chamber; *4* – upper casing; *5* – upper rotor; *6* – rotating piston; *7* – driven shaft; *8* – discharge chamber; *9* – driving shaft; *10* – cavity where the upper rotor piston enters.

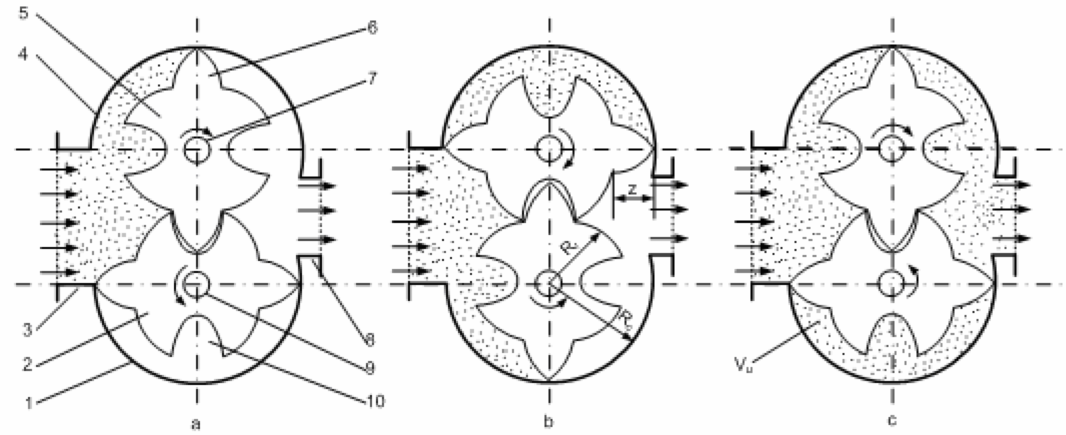

Fig. 3. The functioning principle of the III[rd] version of rotating volumetric machine.
1 – lower casing; *2* – lower rotor; *3* – intake chamber; *4* – upper casing; *5* – upper rotor; *6* – rotating piston; *7* – driven shaft; *8* – discharge chamber; *9* – driving shaft; *10* – cavity where the upper rotor piston enters

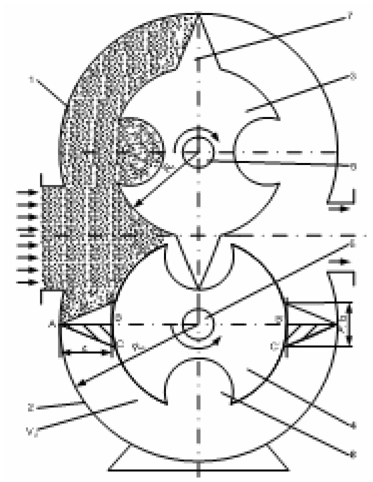

Fig. 4 . Section through the rotating working machine.
1 – upper casing; *2* – lower casing; *3* - upper rotor; *4* - lower rotor; *5,6* – shafts; *7* – triangular piston; *8* – cavity.

A difference between the flow rates corresponding to the I[st] and the II[nd] versions appears:

$$\dot{V}_{p,I} - \dot{V}_{p,II} = 22.042 - 20.422 = 0.1158\ [\mathrm{m^3/h}] \quad (14)$$

For the rotating volumetric compressor with curvilinear shape pistons, the useful volume (V_u) will be reduced with the amount $V_{p,III}$.

It can be noticed from figure 5 that the useful volume V_u will be reduced by a volume equal to the areas (ABC+A\`B\`C\`) multiplied by the lenghth of the piston ($l$), dimension normal to the plane of the figure. But the areas ABC and A\`B\`C\` are equal, thus the volume ($V_{p,III}$) has to be subtracted from V_u:

$$V_{p,III} = 4 \cdot S_{ABC} \cdot l\ [\mathrm{m^3}] \quad (15)$$

Accordingly the flow rate transported by a rotor will decrease with $\dot{V}_{p,III}$:

$$\dot{V}_u = \left[\pi \cdot l \cdot z \cdot \left(z + 2 \cdot R_r\right) - \dot{V}_{p,III}\right]\ [\mathrm{m^3/rot}] \quad (16)$$

Or :

$$\dot{V}_u = \left[\pi \cdot l \cdot z \cdot \left(z + 2 \cdot R_r\right) - \dot{V}_{p,III}\right] \cdot \frac{n}{60}\ [\mathrm{m^3/s}] \quad (17)$$

Because the machine has two rotors, the volumetric flow rate of the machine will be:

$$\dot{V}_{m,III} = 2 \cdot \dot{V}_u = [\pi \cdot l \cdot z \cdot \left(z + 2 \cdot R_r\right) - 4 \cdot S_{ABC} \cdot l] \cdot \frac{n}{30} [m^3/s] \quad (18)$$

The area S_{ABC} of figure 5 is approximated equal to the area S_{ANM} from figure 6.

The area S_{ANM} is enclosed bt the tangent at the rotor in the point N, namely the line MN, and a half of the outline of the piston, namely the curve AM.

The area of the piston section will be:

$$S_p = 2 \cdot S_{ABC} = 2 \cdot S_{ANM} \quad (19)$$

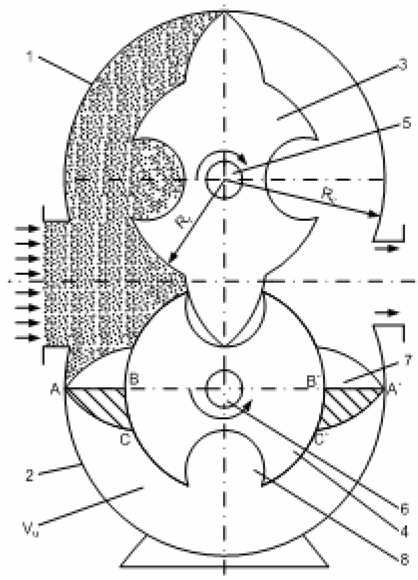

Fig. 5. Transversal section through the rotating working machine.
1 – upper casing; *2* – lower casing; *3* – upper rotor; *4* – lower rotor; *5,6* – shafts; *7* – curvilinear piston; *8* – cavity.

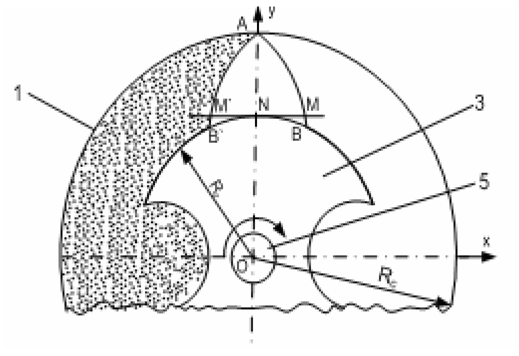

Fig. 6. Section through a rotor with curvilinear shape piston.

The following values were established using a computation software [6] [7]: - The coordinates (x_i;y_i) of the points *A*,*N* and *M*: A(0; 0.008); N(0; 0. 05) and M(0.01929; 0.04613).

The interval *NM* ≈0.02m is divided in 10 intervals of equal length 0.002 m. Therefore the following points on the axis *ox* appear: 0.002; 0.004; 0.006; 0.008; 0.010; 0.012; 0.014; 0.016; 0.018; 0.020 m.

According to these values (x_i), from the table 1, that contains the coordinates of the points situated on the arc AM, the closest values (x_i) are chosen, together with the corresponding values (y_i) [6] [7].

Table 1

Coordinates of the points situated on the arc AM

Nr.	*x* [m]	Nr. punct echivalent	x_i [m]	y_i [m]	$\dot{y}_i$ =y_i-0.5
0≡A	0	-	0	0.00800	0.03000
1	0.002	4	0.01720	0.07876	0.02876
2	0.004	8	0.00406	0.07696	0.02696
3	0.006	12	0.00621	0.07511	0.02511
4	0.008	16	0.00818	0.07323	0.02323
5	0.010	20	0.00998	0.07132	0.02132
6	0.012	25	0.01197	0.06894	0.01894
7	0.014	31	0.01402	0.06611	0.01611
8	0.016	38	0.01595	0.06289	0.01289
9	0.018	49	0.01807	0.05818	0.00818
10	0.020	72	0.01974	0.05022	0.00022

The rotor radius (R_r=0.05m) is subtracted from the values y_i, leading to the values y_i* that decrease from y_i*=0.03 m (point A) to y_i*=0.00022 (point M).

Numerical integration by the method of trapezes was used in order to calculate the area of the curvilinear piston [8] [9]:

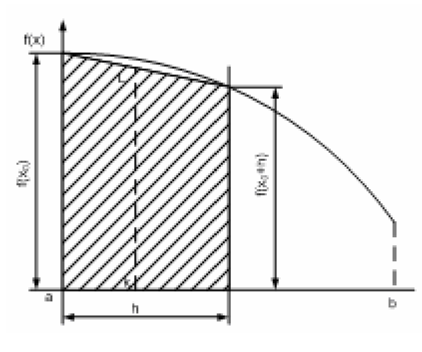

Fig. 7. Computation notations.

The first area is equal to the base (*h*) multiplied by the mean height of the trapeze, namely:

$$KL = \frac{f(x_o)+f(x_o+h)}{2} \tag{20}$$

The area below the function graph will be:

$$I = \int_a^b f(x)\mathrm{d}x \approx \sum_{i=0}^{N-1} \int_{x_i}^{x_{i+1}} f(x)\mathrm{d}x \tag{21}$$

$$I_1 = \frac{h}{2}\cdot\left(y_o+2y_1+2y_2+......2y_{N-1}+y_N\right) \tag{22}$$

Values taken from table I lead to:

$$I_1 = 3.929 \quad [\text{cm}^2] \tag{23}$$

Therefore:

$$S_{ABC}=S_{ANM}= 3.929\cdot10^{-4}\,\text{m}^2.$$

The formula (18) is rewritten (18):

$$\dot{V}_{m,III} = \left[\pi\cdot l\cdot z(z+2\cdot R_r)-4\cdot S_{ABC}\cdot l\right]\cdot\frac{n}{30} \quad [\text{m}^3/\text{s}] \tag{24}$$

If the necessary replacements are made, it results that:

$$\dot{V}_{m,III} = 0.005372 \;\; [\text{m}^3/\text{s}] = 19.213 \;\; [\text{m}^3/\text{h}] \tag{25}$$

In conclusion, the flow rates transported by the machine in the three versions will be:

$$\dot{V}_{M,I} > \dot{V}_{M,II} > \dot{V}_{M,III} \tag{26}$$

$$22.042 > 20.422 > 19.213 \;\; [m^3/h] \tag{27}$$

Or:

$$0.006123 > 0.005673 > 0.005372 \;\; [\text{m}^3/\text{s}] \tag{28}$$

4. CONCLUSIONS

The curvilinear shape of the piston guarantees a better contact between the rotating piston and the cavity of the adjacent rotor (fig.8):

- If the shape of the piston is curvilinear, it can be noticed from figure 8 that there are three contact lines: B, A and C; hereupon the fluid losses between intake and discharge will be reduced, diminishing the „reverse" flow.

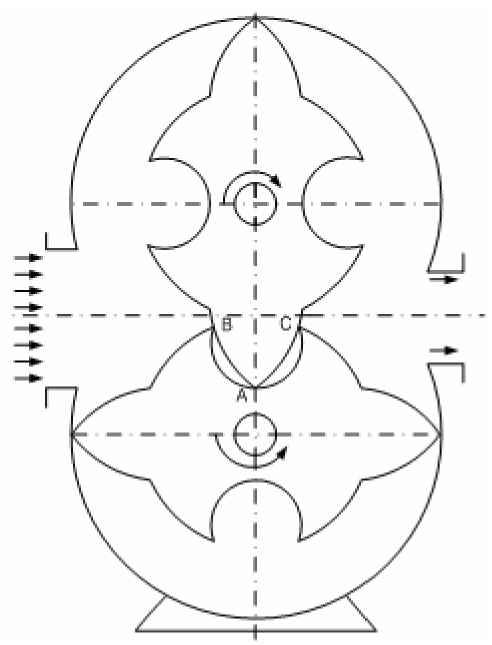

Fig. 8. Transversal section through the rotating machine.

- If the shape of the piston is triangular, there is a sole contact line between the piston and the cavity of the adjacent rotor (zone A).

The essential problem for the rotating working machines with profiled rotors consists in the decrease of the flow rate of the fluid that issues from the „reverse" flow; the radial and front clearances between the piston and the casing and between the two rotors must be hereupon reduced [10] [11].

REFERENCES

[1] **Băran N., Băran Gh.**, "*Steam rotating motor*" Patent no. 111296 released by The State Office for Inventions and Trademarks OSIM Bucharest, **1996**.

[2] **Băran N., Marinescu M., Radcenco V. a.o.** *Technical Thermodynamics, vol I, II, III,* MATRIX ROM Publishing House, Bucharest, **1998**.

[3] **Băran N.**, „*Rotating thermic machines. Working machines. Force machines*" *(in Romanian)*, MATRIXROM Publishing House, Bucharest, **2001**.

[4] **Băran N., Răducanu P., a.o.** – "*Bases of Technical Thermodynamics, Technical Thermodynamics" (in Romanian)*, POLITEHNICA PRESS Publishing House, Bucharest, **2010**.

[5] **Băran N.**, *"Working rotating thermic machines. Machines with profiled rotors. Blade machines" (in Romanian)*, MATRIXROM Publishing House, Bucharest, **2001**.

[6] **Motorga A.**, „*Influence of constructive and functional parameters on the performances of rotating machines with profiled rotors*" *(in Romanian)*, PhD Thesis, Faculty of Mechanical Engineering and Mechatronics, POLITEHNICA University of Bucharest, **2011**.

[7] **Băran N., Motorga A., Costache A.**, "*Computation elements regarding the architecture of a profiled rotor used in the construction of fans*", Termotehnica, no.1, Bucharest, **2008.**

[8] **Bakhvalov N.**"*METHODES NUMERIQUES"* Editions MIR, Moscou , **1976**.

[9] **Postolache M.,** *"Numerical modeling. Theory and applications" (in Romanian)*, pp.59-63, AGIR Publishing House, București, **2008.**

[10] **Cristea V., Crețа G., Ivan D., Ardeleanu P.,** "*Sealings*", Technical Publishing House, Bucharest, **1973**.

[11] **Țurcanu C., Ganea N.**,"*Volumetric Pumps for Liquids*" *(in Romanian)*, Technical Publishing House, Bucharest, **1980**.