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Original Research

Calculation Methods for Volute Parameters Used in the Conceptual Design of Radial and Axial-Centrifugal Compressors

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Abstract

In many designs of a single-stage radial and axial-centrifugal compressors of the turboprop and turboshaft aviation engines, a properly formed collector placed after a vaneless or vaned radial diffuser, is used to decrease velocity and to increase static pressure of an air stream. The spiral diffuser is one of the main diffuser types. A volute is a channel with a different form of transverse sections that gradually expands in the direction of rotor rotation and includes preceding diffuser with a cylindrical inlet hole. Its geometrical parameters should be properly selected to ensure the correct operation of the scroll. This paper presents two main methods of calculation of geometrical parameters of the spiral diffuser: free vortex design (constant angular momentum principle) and constant mean velocity design. Mentioned methods (recommended for use in the conceptual design of a compressor) are based on energy equation - steady flow energy equation, equation of continuity, first law of thermodynamics, Euler's moment of momentum equation, gas dynamics functions and definitions used in theory of turbomachinery. A detailed analysis of geometrical parameters of different types of collectors were conducted. This paper also provides a review of experimental research results of total pressure loss coefficient in the volute and proposed method of determining air stream parameters at volute outlet.

Keywords: turbine engine, compressor, collector

Nomenclature

- a speed of sound
- b passage width of vaneless, vaned diffuser or volute
- c absolute velocity
- c_r radial velocity komponent
- c_u tangential velocity komponent
- D duct diameter of a volume
- F surface area
- F_φ surface area of collector in radial section at angle φ
- h volute Weight
- $\dot{m}$ mass flow rate
- N polytropic exponent/rotational speed
- p static pressure
- r radius of circular volute passage at radial cross-section
- R radius of an infinitesimal section for volute passage measured from impeller axis of rotation
- R_{sr} radius of spiral axis for volute, measured from impeller axis of rotation to centre of gravity of the cross-section
- T static temperature
- α angle of absolute velocity from tangential
- Δ height of threshold
- ρ gas density
- λ Laval number
- φ polar angle (volute azimuth angle) measured from spiral volute tongue towards impeller rotation to the considered radial cross section
- ξ_{sp} total pressure loss coefficient
- ϑ volute sidewalls angle

Subscripts

2 impeller exit
3 vaneless diffuser exit
4 vaned diffuser exit
5 volute exit
6 exit cone discharge condition
$/_{kr}$ critical
$/_{śr}$ mean
$/_{SR}$ centrifugal compressor parameter
$/_{w}$ inner
$/_{z}$ outer
*/ stagnation parameter

1. Introduction

In a centrifugal compressor impeller, an effective work is supplied to the working medium, in which a significant proportion of this work is the increase of kinetic energy of the stream ($c_2 >> c_1$). At the total pressure ratio $\pi_{SR}^* \approx 4$ the absolute velocity at the exit from the impeller reaches the value $C_2 \approx 420$ m/s ($M_{C2} \approx 1{,}1$), whereas for higher values of the stagnation pressure ratio $\pi_{SR}^* \approx 8{,}5$ absolute velocity is greater and is equal to $C_2 \approx 525$ m/s, what corresponds to a Mach number $M_{C2} \approx 1{,}2$. Supersonic flow velocities at the impeller exit require to use efficiently operating diffusers in the design of centrifugal compressor for diffusing the air flow to the velocity necessary for the proper operation of the combustion chamber – usually $C_B \leq 120 \div 160$ m/s ($0{,}1 < M_B \leq 0{,}3$).A spiral diffuser also called collector or volute (Fig.1) is a classic design of a diffuser, which can be found in a number of turboprop engines designs (Allison 250 – B15, Allison 250 – B17) as well as turboshaft engines (GTD – 350, Allison 250 – C20) with mixed-flow compressor, and turboprop (RR500TP) and turboshaft engines equipped with single-stage centrifugal compressor (Allison 250 – C28, Allison 250 – C30, RR300).

Fig. 1. Axial-centrifugal compressor of turboshaft engine GTD-350 - a gift from WSK PZL Rzeszów to the Rzeszów University of Technology: 1 – double jet collector.

In case of low pressure ratio of the centrifugal compressor module $\pi_{SR}^* < 2{,}5$ the volute is preceded by a vaneless diffuser. At higher pressure ratio values $\pi_{SR}^* > 2{,}5$ volute is placed after the vaned diffuser which is preceded by vaneless diffuser (Antas, 2023). In the design practice, there are two main methods used to calculate diffuser's geometrical parameters (Baloni et al., 2012; Ris, 1951; Yahya, 2012):

1) the method based on the law of constant angular momentum (flow without friction – free vortex, constant circulation),
2) the method based on the assumption of constant mean velocity in the volute.

In the methods mentioned above axisymmetrical nature of flow is assumed.

It should be noticed, that in the available literature on the subject lacks publications concerning comprehensive analysis of geometrical parameters of collectors as well as determination of thermal and kinematic parameters of the stream flowing through diffuser (Antas, 2014; Dmitriewskij, 1960; Japikse, 1990, Japoikse, 1996; Reunaunen, 2001; Walczak, 1999; Yahya, 2012).This article is a first publication in the world concerning those problems. According to Van den Braembusshe and Hände (1990) spiral diffuser is probably the most neglected component of the centrifugal compressor in relation to theoretical and experimental studies but, according to the author of this article, the pipe diffuser and controlled –contour diffuser are equally neglected (Antas, 2014, 2019). The geometrical parameters of the volute are (Figs. 2-8):

- inner radius of volute, equal to the outer radius of the preceding diffuser - vaneless: $R_w = R_3 = D_3/2$ or vaned: $R_w = R_4 = D_4/2$,
- outer radius: $R_z = R_z(\varphi) = var$
- mean radius from the impeller axis to the centroid of volute passage: $R_{śr} = R_{śr}(\varphi) = var$.

All above mentioned radii are measured from the impeller axis.

- width or chord of spiral diffuser: $b = idem$ or: $b = b(\varphi) = var$,
- volute height: $h = h(\varphi) = var$.

A volute is a passage with differently shaped cross-sectional areas, which are progressively increasing in the direction of impeller rotation (i.e. diffuser) with a spiral-shaped axis surrounding vaneless gap or vaned diffuser. In practice, most commonly used collectors are (Cumpsty, 1989; Japikse, 1996):

- single jet – a type of a collector, in which single volute surrounds a diffuser along the circumference i.e. at an azimuth angle $\varphi = 2\pi = 360^0$.
- double jet – a kind of volute, in which two collectors are placed on the circumference of a diffuser, each covering half of the circumference at an azimuth angle $\varphi = 2\pi/2 = 180^0$, dividing airflow in the same ratio.

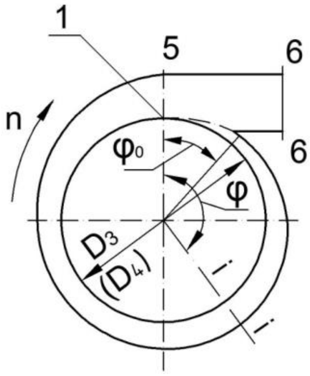

Fig. 2. Single jet scroll (single spiral diffuser).

In Fig.2. and 3. „1" indicates initial segment of a collector located at radius R_3 or R_4, which has coordinate $\varphi = 0$. It corresponds to the collector section with a zero area. Depending on the position of diffuser's wall contour in relation to the axis of vaneless or vaned diffuser, the following volutes can be distinguished (Fig. 4):

- symmetrical,
- asymmetrical.

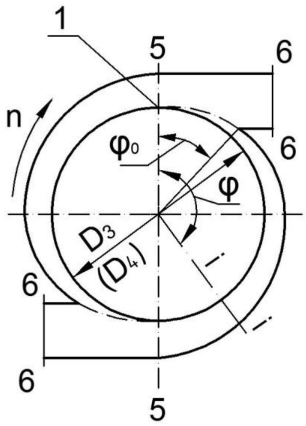

Fig. 3. Double jet collector (double spiral diffuser).

During the design process of collectors the aim is to ensure, that along the inlet section of a collector, the static pressure is approximately constant. Fulfilling the requirement plays an important role in the compressor operation, especially when the volute is preceded by diffuser vane ring or impeller. Otherwise, airflow pulsations will be formed, which can cause flow separation and associated significant losses, and in addition diffuser or impeller blade vibration, threatening the compressor operation reliability (Tuliszka, 1976).

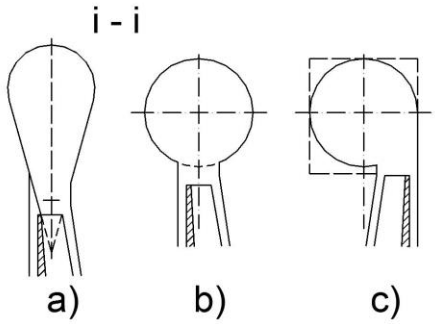

Fig. 4. Shapes of radial sections of the collector (i-i): (a) - symmetrical with an oval section, (b) - symmetrical with a circular section, (c) - asymmetric with a circular or rectangular section.

2. Continuous circulation (free vortex) method

Assuming that flow has axisymmetric nature and is held with no wall friction, then according to Euler's equation the angular momentum is constant, and the change of the absolute velocity in the tangential direction can be determined using a formula:

$$C_u R = C_{4u} R_4 = K = idem \tag{1}$$

if compressor comprises a vaned diffuser - or:

$$C_u R = C_{3u} R_3 = K = idem \tag{2}$$

if compressor comprises only a vaneless diffuser.

The working medium flowing from the vaneless or vaned diffuser is collected in volute starting from cross-section with the coordinate $\varphi = 0$. Through the radial cross-section of the collector (i-i) with the coordinate φ [rad] a mass flow is flown depending on positioning in the compressor:

- when a vaned diffuser is placed before the spiral diffuser

$$\dot{m}_\varphi = \dot{m}\frac{\varphi}{2\pi} = 2\pi R_4 b_4 c_{4r} \rho_4 \frac{\varphi}{2\pi} \tag{3}$$

- or, if a spiral-shaped diffuser is placed after the vaneless diffuser

$$\dot{m}_\varphi = \dot{m}\frac{\varphi}{2\pi} = 2\pi R_3 b_3 c_{3r} \rho_3 \frac{\varphi}{2\pi} \tag{4}$$

We consider the flow through the collector installed directly after the vaneless (slot) diffuser. We take into account different types of volute design.

2.1. Circular volute

The mass flow rate flowing through the radial cross-section of the collector with the coordinate φ can be written using the tangential absolute flow velocity component C_u in the collector, the cross-section area $F = \pi r^2$and the density ρ_3, which is assumed to be constant for the whole section area F. Then, according to the author of monograph (Kholscevnikov, 1970) - (Fig. 5):

$$\dot{m}_\varphi = \rho_3 \iint\limits_F C_u dF = \rho_3 \iint\limits_F C_u R \frac{dF}{R} = \rho_3 C_{3u} R_3 \int_{R_3}^{R_z} \frac{b(r)dR}{R} = K\rho_3 \int_{R_3}^{R_z} \frac{b(r)dR}{R} \tag{5}$$

The equation can be written as:

$$\dot{m}_\varphi = \dot{m}\frac{\varphi}{2\pi} = K\rho_3 \int_{R_3}^{R_z} \frac{b(r)dR}{R} \tag{6}$$

therefore:

$$\varphi = \frac{2\pi}{\dot{m}} K\rho_3 \int_{R_3}^{R_z} \frac{b(r)dR}{R} \tag{7}$$

By denoting the collector cross-section radius with „r", the relation between chord b and radius (Fig.6) can be written as: $\left(b/2\right)^2 + c^2 = r^2$, where: $c = (R - R_3 - r)$, thus:

$$\left(\frac{b}{2}\right)^2 = r^2 - (R - R_3 - r)^2 \tag{8}$$

or:

$$b = 2\sqrt{r^2 - (R - R_3 - r)^2} \tag{9}$$

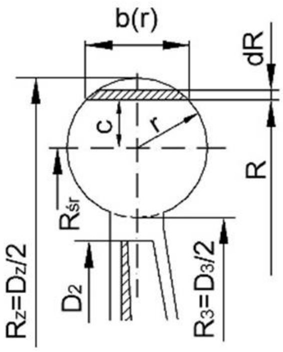

Fig. 5. Schematic designations in the circular collector of the radial compressor: Rśr - radius of the spiral axis, R_3 - inner radius of the collector (R_3 = idem), Rz - outer radius of the collector (Rz = var), b (r) - chord of the collector, r - the collector cross-section radius

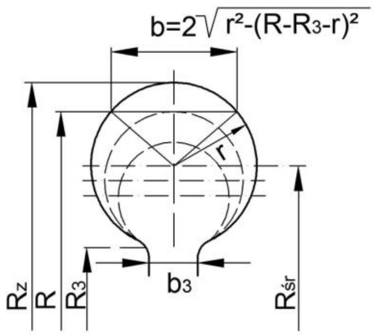

Fig. 6. Spiral collector with a circular section

Therefore:

$$\varphi = \frac{2\pi}{\dot{m}} K\rho_3 \int_{R_3}^{R_z} \frac{2\sqrt{r^2 - (R - R_3 - r)^2}}{R} dR = \frac{2\pi}{\dot{m}} K\rho_3 \int_{R_3}^{R_3+2r} \frac{2\sqrt{r^2 - (R - R_3 - r)^2}}{R} dR = \frac{4\pi^2}{\dot{m}} K\rho_3 \left[(R_3 + r) - \sqrt{(R_3 + r)^2 - r^2}\right] \tag{10}$$

The above equation, obtained as a result of complex equations' transformations, determines relation between the angle φ and the radius r of the circular cross-section, as well as the volute shape. Solving the above quadratic equation in relation to „r" the following relation is obtained:

$$r = \frac{\varphi \dot{m}}{4\pi^2 K\rho_3} + \sqrt{R_3 \frac{\varphi \dot{m}}{2\pi^2 K\rho_3}} \tag{11}$$

By specifying the angle φ, the radius r can be determined (the angle is expressed in [rad]). If the angle φ in the relation (11) is expressed in degrees, then:

$$\varphi^{\circ} = 57.3 \cdot \varphi \tag{12}$$

Whereas, formula (11) for radius of the collector cross-section is:

$$r = \frac{\varphi^{\circ}\ \dot{m}}{4\pi^2 K\rho_3 \cdot 57.3} + \sqrt{R_3 \frac{\varphi^{\circ}\ \dot{m}}{2\pi^2 K\rho_3 \cdot 57.3}} \tag{13}$$

For the following case, if the angle φ in the equation (11) is expressed in radians, the diameter of the circular collector is:

$$D = 2r = \frac{\varphi\dot{m}}{2\pi^2 K\rho_3} + \sqrt{2R_3 \frac{\varphi\dot{m}}{\pi^2 K\rho_3}} \tag{14}$$

While, if the angle φ is expressed in degrees:

$$D = 2r = \frac{\dot{\varphi}^{\circ}\ \dot{m}}{2\pi^2 K\rho_3 \cdot 57.3} + \sqrt{2R_3 \frac{\varphi^{\circ}\ \dot{m}}{\pi^2 K\rho_3 \cdot 57.3}} \tag{15}$$

Collector's external radius in random cross-section (i-i):

$$R_z = R_3 + D \tag{16}$$

If a volute is preceded by a vaned diffuser, then in the above relations the following should be substituted:

$$R_3 \equiv R_4 \tag{17}$$

$$\rho_3 \equiv \rho_4 \tag{18}$$

$$K = C_{4u} R_4 \tag{19}$$

$$\dot{m} = 2\pi R_4 b_4 c_{4r} \rho_4 \tag{20}$$

The radius of the volute's central axis is then:

$$R_{sr} = R_3 + r \tag{21}$$

or if a circular collector is preceded by a vaned diffuser:

$$R_{sr} = R_4 + r \tag{22}$$

2.2. Constant-width volute (rectangular)

In this case $b(r) = b = idem$ (Fig. 7). Therefore equation (7) for the angle φ expressed in radians can be rewritten as:

$$\varphi = \frac{2\pi}{\dot{m}} K\rho_3 \int_{R_3}^{R_z} \frac{b(r)dR}{R} = \frac{2\pi}{\dot{m}} K\rho_3 bR|_{R_3}^{R_z} = \frac{2\pi}{\dot{m}} K\rho_3 b(\ln R_z - \ln R_3) = \frac{2\pi}{\dot{m}} K\rho_3 b \ln R\ \frac{R_z}{R_3} \tag{23}$$

Therefore:

$$\frac{R_z}{R_3} = e^{\frac{\dot{m}\varphi}{2\pi K b \rho_3}} \tag{24}$$

and finally:

$$R_z = R_3 e^{\frac{\dot{m}\varphi}{2\pi K b \rho_3}} \tag{25}$$

In this case the curve that determines external contour of spiral diffuser is a logarithmic spiral.

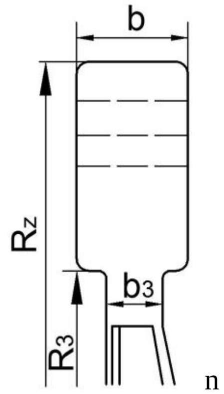

Fig. 7. Spiral collector with a rectangular cross-section.

According to equation (23):

$$\varphi = \frac{2\pi}{\dot{m}} K \rho_3 b \ln \frac{R_z}{R_3}$$

and using the formula (2) (or (1) and from the relation:

$$\dot{m} = 2\pi R_3 b_3 C_{3r} \rho_3 \tag{26}$$

or the relationship (20):

$$\dot{m} = 2\pi R_4 b_4 C_{4r} \rho_4$$

this gives:

$$\varphi = \frac{2\pi R_3 b C_{3u} \rho_3}{2\pi R_3 b_3 C_{3r} \rho_3} ln \frac{R_z}{R_3} = \frac{C_{3u} b}{C_{3r} b_3} ln \frac{R_z}{R_3} \tag{27}$$

but:

$$\ln x \cong 2.3 \log x \tag{28}$$

therefore:

$$\varphi \cong 2.3 \frac{C_{3u} b}{C_{3r} b_3} log \frac{R_z}{R_3} \tag{29}$$

or in degrees:

$$\varphi^{\circ} = 57.3 \cdot \varphi \cong 132 \frac{C_{3u} b}{C_{3r} b_3} log \frac{R_z}{R_3} \tag{30}$$

If a spiral volute is preceded by a vaned diffuser the following relations are received:

$$R_z = R_4 e^{\frac{\dot{m}\varphi}{2\pi K b \rho_4}} \tag{31}$$

where: $K = C_{4u} R_4$, $\dot{m} = 2\pi R_4 b_4 C_{4r} \rho_4$

and:

$$\varphi = \frac{2\pi}{\dot{m}} K \rho_4 b \ln \frac{R_z}{R_4} \tag{32}$$

$$\varphi = \frac{C_{4u} b}{C_{4r} b_4} \ln \frac{R_z}{R_4} \tag{33}$$

$$\varphi \cong 2{,}3 \frac{C_{4u} b}{C_{4r} b_4} \log \frac{R_z}{R_4} \tag{34}$$

$$\varphi^{\circ} = 132 \frac{C_{4u} b}{C_{4r} b_4} log \frac{R_z}{R_4} \tag{35}$$

The equation (25) can be simplified by deployment of it in series, and then:

$$R_z = R_3 \left[1 + \frac{\dot{m}}{2\pi K b \rho_3} \varphi + \left(\frac{\dot{m}}{2\pi K b \rho_3} \right)^2 \frac{\varphi^2}{2} + \cdots \right] \tag{36}$$

and omitting expressions in the power factor greater than one (higher order negligible):

$$R_z = R_3 + R_3 \frac{\dot{m}}{2\pi K b \rho_3} \varphi \tag{37}$$

or taking into account formulas (2) and (26):

$$R_z = R_3 + R_3 \frac{C_{3r} b}{C_{3u} b_3} \varphi \tag{38}$$

then:

$$\varphi \cong \frac{R_z - R_3}{R_3} \frac{C_{3u} b}{C_{3r} b_3} \tag{39}$$

or expressed in degrees:

$$\varphi^{\circ} = 57.3 \frac{R_z - R_3}{R_3} \frac{C_{3u} b}{C_{3r} b_3} \tag{40}$$

The equation (37) gives less accurate results and presents Archimedean spiral. The radius of the central axis of the rectangular cross-section volute is:

$$R_{sr} = R_3 + \frac{R_z - R_3}{2} = \frac{R_z + R_3}{2} \tag{41}$$

2.3. Square volute

For a volute with square cross-section (Fig. 8) the following relation can be written:

$$b = R_Z - R_3 \tag{42}$$

Because the angle φ is determined by the formula (7):

$$\varphi = \frac{2\pi}{\dot{m}} K\rho_3 \int_{R_3}^{R_z} \frac{b(r)dR}{R}$$

therefore after taking into account the relation *b(r) = b = idem* the following relation – as for a rectangular volute – is obtained (23):

$$\varphi = \frac{2\pi}{\dot{m}} K\rho_3 b \ln\frac{R_z}{R_3}$$

And after using the formula (42):

$$\varphi = \frac{2\pi}{\dot{m}} K\rho_3 (R_z - R_3) \ln\frac{R_z}{R_3} \tag{43}$$

or in degrees, applying the relation for a rectangular volute (35):

$$\varphi^\circ = 132\frac{C_{3u}b}{C_{3r}b_3}\log\frac{R_z}{R_3}$$

therefore using the relation (42):

$$\varphi^\circ = 132\frac{C_{3u}(R_z - R_3)}{C_{3r}b_3}\log\frac{R_z}{R_3} \tag{44}$$

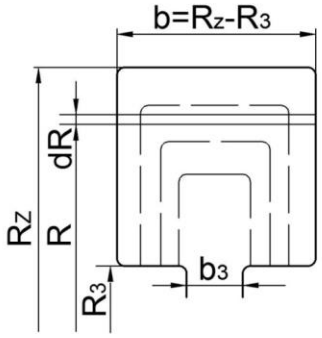

Fig. 8. Spiral collector with a square cross-section.

The volute passage width (height) $b = R_z - R_3$ – eq. (42) can be determined by integrating at the volute discharge (exit) cross-section ($b = b_{wyl} = b_5$). Assuming that in this cross-section on the radius of the volute spiral axis

$$R_{5sr} = R_3 + b/2 \tag{45}$$

the flowing airflow reaches the following velocity:

$$C_{5usr} = \frac{C_{3u}R_3}{R_{5sr}} \tag{46}$$

or generally:

$$C_u = \frac{C_{3u}R_3}{R} \tag{47}$$

while infinitesimal mass flow rate:

$$d\dot{m} = \rho_3 C_u b dR \tag{48}$$

therefore:

$$\dot{m} = \rho_3 \int_{R_3}^{R_z} C_u b dR = \rho_3 \int_{R_3}^{R_3+b} C_u b R_3 \frac{dR}{R} \tag{49}$$

then:

$$\dot{m} = \rho_3 b C_{3u} R_3 \ln \frac{R_3 + b}{R_3} \quad or \quad \dot{m} = K\rho_3 b \ln \frac{R_3 + b}{R_3} \tag{50}$$

This equation is solved using the method of successive approximations. The value of the left side of the equation is known and is written by the equation (26). The width or height of volute's passage with square cross-section can also be approximated. Taking into account, that in the spiral volute cross-section (5-5) in its axis, the flowing airflow velocity is determined by the relation (46), then after using this formula we receive:

$$C_{5usr} = \frac{C_{3u} R_3}{R_3 + \frac{b}{2}} \tag{51}$$

The mass airflow rate flowing through this cross-section (5-5) is accordingly ($\rho_{\,5sr} = \rho_3$):

$$\dot{m} = \dot{m}_5 = \rho_3 C_{5usr} F_5 \tag{52}$$

therefore:

$$\dot{m} = \rho_3 \frac{C_{3u} R_3}{R_3 + \frac{b}{2}} b^2 \tag{53}$$

On the other hand, it is known that mass flow rate is determined by the formula (26), then:

$$2\pi R_3 b_3 C_{3r} = \frac{C_{3u} R_3}{R_3 + \frac{b}{2}} b^2 \tag{54}$$

Solving the above quadratic equation with respect to „b" the following is obtained:

$$b = \frac{\pi}{2} b_3 \frac{C_{3r}}{C_{3u}} + \sqrt{\left(\frac{\pi b_3\, c_{3r}}{2c_{3u}}\right)^2 + \left(\frac{\pi b_3\, c_{3r}}{c_{3u}}\right)} \tag{55}$$

Note: In case of a single jet volute in above relations it should be considered, that: $\dot{m} = \dot{m}\,; \varphi = 0 \ldots 2\pi\,;$ or $\varphi^{\circ} = 0 \ldots 360°$, whereas for a double spiral diffuser: $\dot{m} = \frac{\dot{m}}{2}\,; \varphi = 0 \ldots \pi\,;$ or $\varphi^{\circ} = 0 \ldots 180°$.

3. Constant mean velocity method

The second type of method for calculating the geometrical parameters of volutes is based on the assumption of the mean velocity flow value in the considered spiral diffuser cross-section (Ris, 1951; Yahya, 2012).

3.1. Trapezoidal or oval cross-section volute (pear-shaped)

By determining the volute cross-section corresponding to the angle φ expressed in radians by F_{φ}, then assuming mean airflow velocity value in this cross-section, for single-jet volute the following relation is obtained:

$$F_{\varphi} = \frac{\dot{m}_{\varphi}}{\rho_3 \cdot c_{sr}} \tag{56}$$

or:

$$F_{\varphi} = \frac{\dot{m}\varphi^{\circ}}{\rho_3 \cdot 360 \cdot c_{sr}} \tag{57}$$

while, for a double jet collector this formula can be written as:

$$F_{\varphi} = \frac{\dot{m}\varphi^{\circ}}{\rho_3 \cdot 180 \cdot c_{sr}} \tag{58}$$

Expressing the surface area of a collector in radial section by F_{φ} as function of given geometrical parameters i.e. $F_{\varphi} = f(b_{3,} \vartheta, h)$ – for the trapezoidal volute and $F_{\varphi} = f(b_{3,} \vartheta, h_c))$ – for the oval volute, the following h and h_c parameters respectively can be determined from the continuity equation (Fig. 9).

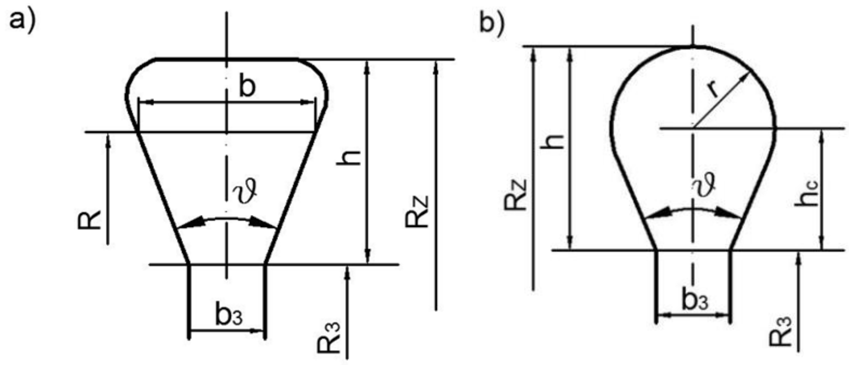

Fig. 9. Schematic designations in the trapezoidal (a) and oval (b) collector: h - the volute height, ϑ – the divergence angle of the lateral walls of the volute.

In case of trapezoidal scroll cross-section, the volute's height h = R_z–R_3 , corresponding to the angle φ, is related to the radial section area F_{φ} by a quadratic equation:

$$F_{\varphi} = \; tg\frac{\vartheta}{2}h^2 + \; b_3 \cdot h \tag{59}$$

hence:

$$h = \; -\frac{b_3}{2\,tg\frac{\vartheta}{2}} + \sqrt{\left(\frac{b_3}{2\,tg\frac{\vartheta}{2}}\right)^2 + \frac{F_{\varphi}}{tg\frac{\vartheta}{2}}} \tag{60}$$

then, for a single-jet collector the following relation is obtained:

$$h = \; -\frac{b_3}{2\,tg\frac{\vartheta}{2}} + \sqrt{\left(\frac{b_3}{2\,tg\frac{\vartheta}{2}}\right)^2 + \frac{\dot{m}\varphi^{\circ}}{\rho_3 \cdot 360 \cdot c_{sr} \cdot tg\frac{\vartheta}{2}}} \tag{61}$$

or:

$$h = \; -K_1 + \sqrt{K_1^2 + \frac{K_2\,\varphi^{\circ}}{c_{sr}}} \tag{62}$$

where:

$$K_1 = \frac{b_3}{2\,tg\frac{\vartheta}{2}} \tag{63}$$

$$K_2 = \frac{\dot{m}}{\rho_3 \cdot 360 \cdot tg\frac{\vartheta}{2}} \tag{64}$$

The corrective coefficient $K_F = 1.05 \div 1.1$ (Ris, 1951), shall be inserted in order to take into consideration decreasing of volute's cross-section surface area, as a result of corners rounding, so then:

$$K_2 = \frac{\dot{m}K_F}{\rho_3 \cdot 360 \cdot tg\frac{\vartheta}{2}} \tag{65}$$

Width of the spiral diffuser with a trapezoid contour for any radius R value of cross-section can be determined by geometrical relation:

$$b = b_3 + 2(R - R_3)\,tg\frac{\vartheta}{2} \tag{66}$$

For an oval spiral diffuser (Fig. 9b) the distance value h_c if the center of the circle with radius r from inlet radius R_3, corresponding to the polar angle φ, is related to the cross-section of the volute passage F_φ by the following quadratic equation:

$$F_\varphi = \left(\frac{\pi}{2}tg^2\frac{\vartheta}{2} + tg\frac{\vartheta}{2}\right)h_c^2 + \left(\frac{\pi}{2}b_3\,tg\frac{\vartheta}{2} + b_3\right)h_c + \frac{\pi}{8}b_3^2 \tag{67}$$

After solving the above quadratic equation and transformations the following is obtained (Podobuiev & Selezniev, 1957):

$$h_c = -K_1 + \sqrt{\frac{K_1^2 tg\frac{\vartheta}{2} + F_\varphi}{K_3}} \tag{68}$$

or, having regard to the relation F_φ for a single-jet volute, this gives:

$$h_c = -K_1 + \sqrt{\frac{K_1^2 + \frac{K_2'\varphi^\circ}{c_{sr}}}{K_3}} \tag{69}$$

where:

$$K_1 = \frac{b_3}{2\,tg\frac{\vartheta}{2}} \tag{70}$$

$$K_2' = \frac{\dot{m}}{\rho_3 \cdot 360} \tag{71}$$

$$K_3 = \frac{\pi}{2}tg^2\frac{\vartheta}{2} + tg\frac{\vartheta}{2} \tag{72}$$

The other geometric parameters of the volute are determined by relations:

$$r = (h_c + K_1)\,tg\frac{\vartheta}{2} \tag{73}$$

$$h = h_c + r \tag{74}$$

$$R_z = R_3 + h \tag{75}$$

3.2. Circular cross-section volute

For any cross-section of a single-jet volute the section is:

$$F_\varphi = \frac{\pi D^2}{4} \tag{76}$$

then according to the formula (56):

$$\frac{\pi D^2}{4} = \frac{\dot{m}_\varphi}{\rho_3 \ c_{sr}} \tag{77}$$

hence, the volute diameter is:

$$D = 2\sqrt{\frac{\dot{m}_\varphi}{\pi \rho_3 \ c_{sr}}} \tag{78}$$

or according to the equation (57):

$$\frac{\pi D^2}{4} = \frac{\dot{m}\varphi^\circ}{\rho_3 \ c_{sr} 360} \tag{79}$$

whence:

$$D = \sqrt{\frac{\dot{m}\varphi^\circ}{90\pi \rho_3 \ c_{sr}}} \tag{80}$$

And for a double jet volute according to the relation (58):

$$D = \sqrt{\frac{\dot{m}\varphi^\circ}{45\pi \rho_3 \ c_{sr}}} \tag{81}$$

In case of a volute installed at the vaneless diffuser exit a constant value of the average airflow velocity can be assumed along the whole scroll $c_{sr} = (0.7 \div 0.8)c_3$.The divergence angle of the lateral walls recommended for trapezoidal and oval collector is between $\vartheta = 45 \div 60°$ (Podobuiev & Selezniev, 1976; Ris, 1951).

4. Flow parameters at the volute exit

The stagnation pressure loss at the volute depends on placing the spiral diffuser in the compresor passage, therefore depending on the design type, it is determined from the following relations:

- for a volute installed at the impeller exit (compressor without vaneless and vaned diffuser):

$$\Delta p_{2,5}^* = \xi_{sp} \frac{\rho_2 \cdot c_2^2}{2} \tag{82}$$

- for a volute preceded by a vaneless diffuser:

$$\Delta p_{3,5}^* = \xi_{sp} \frac{\rho_3 \cdot c_3^2}{2} \tag{83}$$

- for a volute installed at a vaned diffuser exit:

$$\Delta p_{4,5}^{*} = \xi_{sp} \frac{\rho_4 \cdot c_4^2}{2} \tag{84}$$

The issue of determining the total pressure loss coefficient in the spiral diffuser ξ_{sp}, experimentally determined, was the subject of consideration only in a few papers (Abduraszików et al., 1974; Bajle, 1981; Biełousov et al., 2003; Eckert, 1959; Japikse, 1990; Tuliszka, 1976; Walczak, 1999; Witkowski, 2004). However, this issue can avoided, or at least the uncertainty reduced, by using numerical simulations to estimate losses for the considered configuration.

According to Abduraszików et al. (1974) the total pressure loss coefficient value in a volute (Fig. 10) depends on the location of a spiral diffuser in a compressor passage and on the absolute velocity angle of α (α_2, α_3) at collector inlet. Due to high values of the total pressure loss coefficient in the centrifugal compressors of turbine engines, the volute is not installed directly behind the rotor exit.

The most complex analysis of total pressure loss coefficient changes in a spiral diffuser as a function of the angle of absolute velocity of stream outflow from vaneless diffuser α_3 was introduced in the paper by Walczak (1999) – Fig.11. This angle value is related with compressor's operating point from surge mass flow rate ($\dot{m}_{min}$) to maximum mass flow rate ($\dot{m}_{max}$).

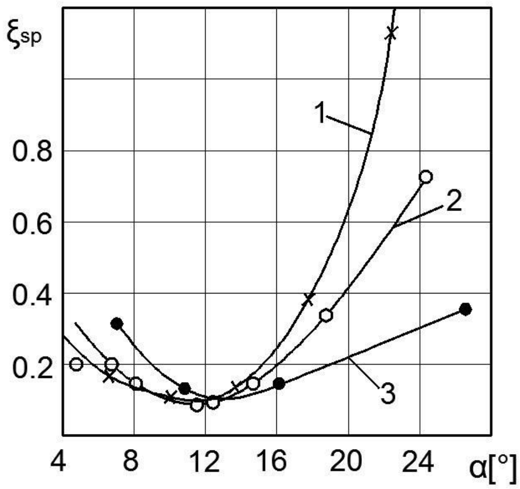

Fig. 10. Changes of the total pressure loss coefficient in the collector vs. angle of absolute velocity at the collector inlet. 1, 2 - the volute mounted at the outlet of the impeller , 3 - the volute preceded by a vaneless diffuser.

The analysis shown in the Fig. 10 and Fig. 11 show that discrepancies between the results of the research of individual authors are considerable. In order to avoid doubts regarding to the reliability of the research results, it is recommended to conduct extensive additional experimental researches based on measurements of pressure, temperature and velocity at volute's inlet and exit cross-section, avoiding completing the researches with velocity calculations in control cross-sections.

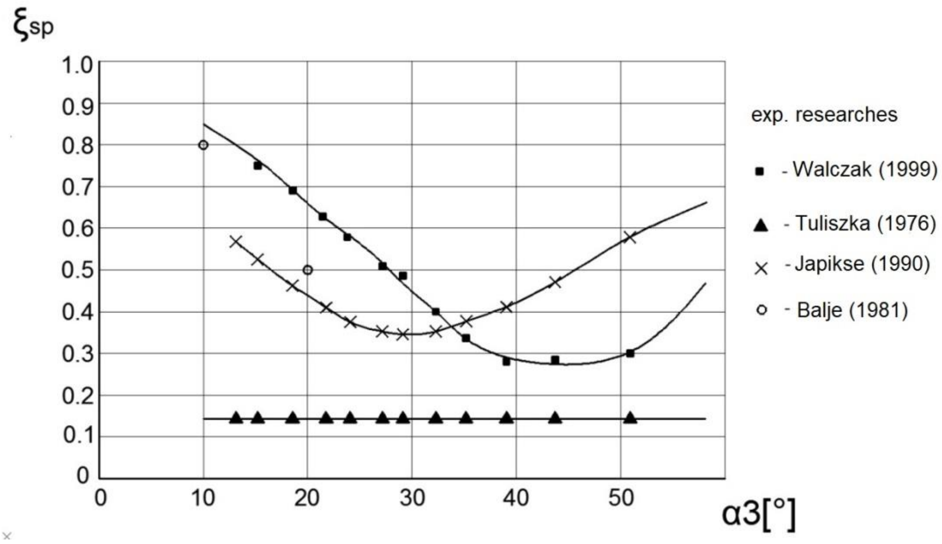

Fig. 11. The variation of the total pressure loss coefficient in the collector vs. angle of absolute velocity α_3 at the collector inlet (Walczak, 1999).

According to Tuliszka (1976) the total pressure loss coefficient in a spiral diffuser has a constant value regardless of the angle α_3, and its value is definitely lower. Similar recommendations are given in the paper by Biełousov et al. (2003) by, while (ξ_{sp}=0.06÷0.15) and in the monograph by Witkowski (2004) (where ξ_{sp}=0.22).

- Stagnation pressure p_5^* at the volute exit cross-section (5-5) is defined according to the design of a centrifugal compressor. Thus, in the case of collector placed at impeller's exit and behind a vaneless diffuser the following is obtained:

$$p_5^* = p_2^* - \Delta p_{2,5}^* \tag{85}$$

$$p_5^* = p_3^* - \Delta p_{3,5}^* \tag{86}$$

while, in case of the collector preceded by a vaned diffuser:

$$p_5^* = p_4^* - \Delta p_{4,5}^* \tag{87}$$

- The stagnation temperature at the exit cross-section is:

$$T_5^* = T_2^* = T_3^* = T_4^* \tag{88}$$

- The absolute airflow velocity at the exit cross-section of a volute preceded by a vaneless diffuser is given by the equation (46):

$$c_5 = c_{5usr} = \frac{c_{3u} \cdot R_3}{R_{sr5}}$$

- The critical speed of sound:

$$a_{kr5} = \sqrt{\frac{2 \cdot k \cdot R}{k+1} T_5^*} \tag{89}$$

- Laval number of the absolute velocity for stream outflow from the volute:

$$\lambda_5 = \frac{c_5}{a_{kr5}} \tag{90}$$

- Gasdynamics function of pressure:

$$\Pi(\lambda_5) = (1 - \frac{k-1}{k+1} \lambda_5^2)^{\frac{k}{k-1}} \tag{91}$$

- Static pressure:

$$p_5 \equiv p_{\varphi=2\pi} = \Pi(\lambda_5) \cdot p_5^* \tag{92}$$

- Static temperature:

$$T_5 \equiv T_{\varphi=2\pi} = T_3^* - \frac{c_5^2}{\frac{2 \cdot k \cdot R}{k+1}} \tag{93}$$

or:

$$T_5 = \tau(\lambda_5) \cdot T_5^* \tag{94}$$

where, gasdynamics function of temperature

$$\tau(\lambda_5) = 1 - \frac{k-1}{k+1}\lambda_5^2 \tag{95}$$

Friction work losses in a volute preceded by vaneless diffuser

$$l_{r3,5} = \xi_{sp}\frac{c_3^2}{2} \tag{96}$$

- The polytrophic exponent of the compression in a volute n ≡ $n_{3,5}$ is determined from the relation (First Law of Thermodynamics):

$$l_{r3,5} = \left(\frac{k}{k-1} - \frac{n}{n-1}\right)R(T_5 - T_3) \tag{97}$$

or from the polytrophic process equation:

$$\frac{n}{n-1} = \frac{ln\frac{p_5}{p_3}}{ln\frac{T_5}{T_3}} \tag{98}$$

usually: n=1.9÷2.0

- The single jet volute's exit radius – for φ = 2π from the equation (11) results from:

$$r_5 = \frac{\dot{m}}{2\pi \cdot K \cdot \rho_3} + \sqrt{\frac{R_3 \cdot \dot{m}}{\pi \cdot K \cdot \rho_3}} \tag{99}$$

- The mean volute exit radius:

$$R_{sr5} = R_3 + r_5 \tag{100}$$

Checking the value of the single jet circular volute exit radius could be performed in the following sequence:

- Static pressure:

$$p_5 = p_3\left(\frac{T_5}{T_3}\right)^{\frac{n}{n-1}} \tag{101}$$

- Ratio of stagnation pressures at the collector:

$$\sigma_{3,5} = 1 - \xi_{sp}\frac{k \cdot \lambda_3^2}{k+1}(1 - \frac{k-1}{k+1}\lambda_3^2)^{\frac{k}{k-1}} \tag{102}$$

- Stagnation pressure:

$$p_5^* = \sigma_{3,5} \cdot p_3^* \tag{103}$$

- Mass flux density gasdynamics function:

$$q(\lambda_5) = \lambda_5(1 - \frac{k-1}{k+1}\lambda_5^2)^{\frac{k}{k-1}}(\frac{k+1}{2})^{\frac{k}{k-1}} \tag{104}$$

- Surface area of the volute exit:

$$F_5 = \frac{\dot{m}\sqrt{T_5^*}}{s \cdot p_5^* \cdot q(\lambda_5)} \tag{105}$$

where constant in the continuity equation for air: $s = 0{,}0404(\frac{J}{kgK})^{-0.5}$

- Single jet volute exit radius:

$$r_5 = \sqrt{\frac{F_5}{\pi}} \tag{106}$$

5. Discussion

The considerations presented in this paper regarding selection of cross section areas $F = F(\varphi)$ for typical scroll shapes were conducted on the basis of the inviscid medium model – equation (5). However the losses occurring in the volute cause corresponding pressure decreases. According to Eckert (1959) it is recommended to apply additional delay of the medium, that is additional increase of radial section F, so that in this manner take into consideration the influence of friction and separation in the volute tongue area (from $\varphi = 0\ to\ \varphi = \varphi_0$).

However, there is no uniform view on the method of volute design presented above, as e.g. studies by Reunanen (2001) on spiral diffusers show that the volutes designed on the basis of inviscid medium model operate in the best way. Therefore it seems to be a little safer to adopt collector design principle on the basis of the equation (7) and not to introduce corrections resulting from pressure losses, although the method of B. Eckert is also allowable and appropriate. It is recommended to compile calculated volute parameters in the aggregate chart.

The flow through spiral diffuser can be treated to some extent as a flow through curved passage. As with any type of flow as in this passage, secondary flows arise in it (induced, knee flow), marked schematically in the Fig. 12. The flow through these diffusers was examined experimentally by Krantz using a fluid as a working medium. As a result of these researches it was stated, that particle tracks are not spiral, but helical. This causes, that the path by which working medium is flowing is several times greater than volute axis distance (Hariharan & Govardhan, 2015; Tuliszka, 1976).

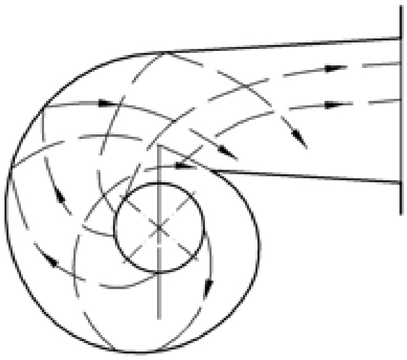

Fig. 12. Medium flow through a collector (Cumpsty, 1989).

In case of symmetrical collectors the introduction of a symmetrical flow leaving a vaneless or vaned diffuser, so heading toward the center line, intensifies the secondary flow and simultaneously increases losses. If the stream outflowing from the diffuser is introduced tangentially to the volute – Fig. 13. – then with unilaterally formed spiral diffuser, the double vortex can be reduced to single vortex, which causes smaller losses (Cumpsty, 1989).

Centrifugal compressor stage characteristics with three different volutes formed symmetrically are shown in the Figure 14 (symmetrical spiral diffuser a and b) and asymmetrical (volute c). Escher-Wyss researches show that volutes with asymmetrical stream inlet operates with lower losses, therefore this type of design, if there are no other design or technological considerations, is recommended for centrifugal compressors.

The problem of asymmetrical delivery of working fluid to the collector was subject of few theoretical and experimental researches. According to Reunanen (2001) the shape of volute passage cross-section satisfying the condition of symmetrical and asymmetrical working fluid delivery are equivalent in terms of operation of the centrifugal compressor. It should be noticed, that asymmetrical volute in case of rectangular or square contour (Fig. 15) does not create additional designing problems, where as in a circular-shaped diffuser, asymmetrical working fluid delivery causes creation of threshold of height Δ – Fig. 15.b (Cumpsty, 1989; Pan et al., 1999).

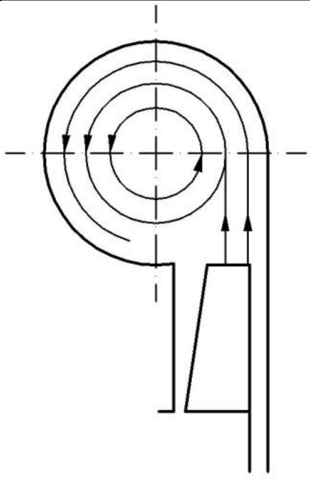

Fig. 13. The stream lines in the cross-section of the asymmetrical collector (Tuliszka, 1976).

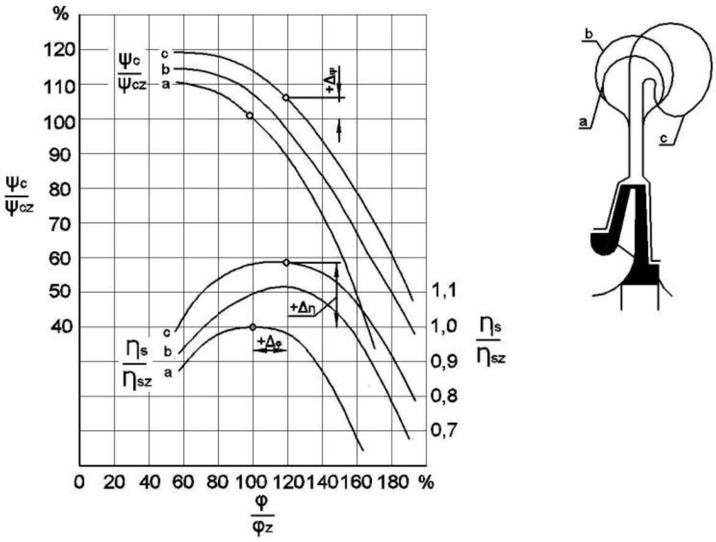

Fig. 14. Effect of the volute shape on characteristics and efficiency of the radial compressor stage (test results of Escher-Wyss Zurich); ψ_c - the pressure rise coefficient, η_S - isentropic efficiency, φ – flow coefficient (subscript z refers to the nominal operating conditions) (Eckert, 1959).

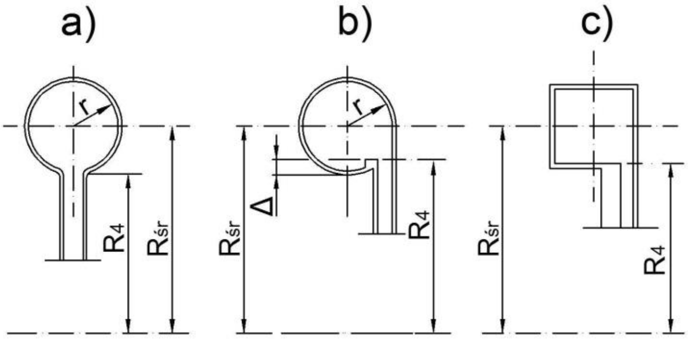

Fig. 15. Designation of geometrical parameters of the volute with a circular contour: a) symmetric, b) asymmetric and c) asymmetric with a rectangular section.

The radius from impeller axis to the centroid of the symmetrical volute with a circular contour, preceded by vaned diffuser in accordance with formula (14) is $R_{sr} = R_4 + r$, therefore the difference:

$$R_{sr} - r = R_4 \tag{107}$$

has a constant value.

In case of asymmetrical spiral diffuser with a circular cross-section the above equation is not fulfilled, because the threshold height is:

$$\Delta = R_4 - (R_{sr} - r) \tag{108}$$

whereas according to research conducted by Pan et al. (1999) the difference ($R_{śr} - r$) is linear function of angle φ° expressed in degrees, i.e.:

$$(R_{sr} - r) = a + b\varphi^o \tag{109}$$

where constants: a = 0.0823 m, b = 2.33 10^{-5} $m/^0$.

The design of a compressor with specific technical conditions can be divided into three stages. The first stage, covered by the conceptual design, includes determining the number of stages and their load, appropriate selection of speed at a set radius in individual stages, determining the flow channel in a meridional cross-section, assessing the density and profiling of the compressor rim. Many processes cannot be explained using one- and two-dimensional models, but such models, together with the cylindrical flow model, allow obtaining flow equations that, despite numerous simplifications, provide sufficient accuracy for the initial quantitative analysis of flows in aircraft turbomachines. These methods are commonly used to determine the initial geometry of the flow channel of turbomachines at the stage of design assumptions. The second stage, which is the preliminary design of the assembly, includes a more accurate calculation of the flow kinematics and the parameters of the thermodynamic state in the gaps between the rims. In these calculations, an axisymmetric model of gas flow through the turbomachine rims is usually assumed. The third stage, covered by the technical design, includes a detailed quantitative analysis of the flow of the medium through the individual rims of the fluid-flow machine stages. In the calculations for the technical design, it is recommended to use two-dimensional and three-dimensional flow models of the viscous or inviscid medium. The Navier-Stokes or Euler equations are used for quantitative analysis. It is advisable to analyse both a two- and three-dimensional flow models and to use them in the order of increasing complexity.

6. Summary

Although experimental studies of spirals of different geometry were performed as early as the 1940s, these studies were limited due to high production costs. In recent years, numerical simulations have provided significant progress in the analysis of spirals of different designs. An example is the work of Hottois et al. (2023), which presents an adjoint-based optimization of a volute of the SRV2-O compressor based on computational fluid dynamics modeling. Due to the high instrumentation costs, experimental analysis of volute is the domain of large industrial companies, which do not provide research results due to legal regulations. For this reason, the total number of studies in the open literature concerning the volute geometry is small (Heinrich & Schwarze, 2016). The shape optimization of radial compressors mainly focuses on improving the impeller. However, as indicated by Hottois et al. (2023) the volute plays a key role in the overall performance of the compressor.

The geometrical parameters of the volute should be properly selected to ensure the correct operation of the diffuser. The constant mean velocity design and free vortex design presented in this work are recommended for use in the conceptual design of a compressors. The selection of the cross-section of typical scroll shapes presented in this article was based on the inviscid medium model. A detailed analysis of geometrical parameters of different types of collectors and proposed method of determining air stream parameters at volume outlet may be helpful for compressor designers.

References

Abdurasziłow, S. A. et al. (1974). *Pumps and compressors*. Niedra (in Russian).

Antas, S. (2014). Pipe diffuser for radial and axial-centrifugal compressor. *International Journal of Turbo & Jet-Engines*, 31(1), 29-36. https://doi.org/10.1515/tjj-2013-0031

Antas, S. (2019). Exhaust system for radial and axial-centrifugal compressos with pipe diffuser. *International Journal of Turbo & Jet-Engines*, *36*(3), 297-304. https://doi.org/10.1515/tjj-2016-0068

Antas, S. (2023). *Theory of turbomachines for aviation turbine engines*. Publishing House of Rzeszów University of Technology (in Polish).

Aungier, R. H. (2000). *Centrifugal compressors*. ASME Press.

Balje, O. E. (1981). *Turbomachines. A guide to design selection and theory*. Wiley-Interscience.

Baloni, B. D., & Channiwala, S. A. (2013). Design and analysis of volute casing: A review. In *Proceedings of ASME Turbo Expo 2012: Turbine Technical Conference and Exposition*, paper no. GT 2012-68056 (pp. 657-664). https://doi.org/10.1115/GT2012-68056

Baloni, B. D., Channiwala, S. A., & Mayavanshi, V. K. (2012). Pressure recovery and loss coefficient variations in the two different centrifugal blower volute designs. *Applied Energy*, *90*, 335-343. https://doi.org/10.1016/j.apenergy.2011.02.016

Biełousov, A. N., Musatkin, N. F., & Radko, W. M. (2003). *Theory and calculations of aircraft turbo-machines*. Samarski Gosudarstvenni Aerokosmiceski Institut (in Russian).

Cumpsty, N. A. (1989). *Compressor aerodynamics*. Longman Scientifics and Technical.

Dmitriewskij, W. I. (1960). *Gasdynamics calculations and design of radial compressor stage*. Technical report No. 137. CIAM (in Russian).

Eckert, B. (1959). *Axial and radial compressors. Application, theory, calculation*. PWT (in Polish).

Hariharan, C., & Govardhan, M. (2015). Effect of inlet clearance on the aerodynamic performance of centrifugal blower. *International Journal of Turbo & Jet-Engines*, *33*, 215-228. https://doi.org/10.1515/tjj-2015-0026

Heinrich, M., & Schwarze, R. (2016). Genetic algorithm optimization of the volute shape of a centrifugal compressor. International Journal of Rotating Machinery, Article 4849025. https://doi.org/10.1155/2016/4849025

Hottois, R., Châtel, A., & Verstraete, T. (2023). Adjoint-based design optimization of a volute for a radial compressor. *International Journal of Turbomachinery Propulsion and Power*, *8*(4), Article 41. https://doi.org/10.3390/ijtpp8040041

Japikse, D. (1990). *Centrifugal compressor design and performance*. Concepts ETI Inc. Norwich, Vermont, Course held at Borsing GmbH Berlin.

Japikse, D. (1996). *Centrifugal compressor design and performance*. Concepts ETI Inc.

Kholscevnikov, K. V. (1970). *Theory and calculations of aircraft turbo-machines*. Masinostrostrojenije (in Russian).

Pan, D., Whitfield, A., & Wilson, M. (1999). Design considerations for the volutes of centrifugal fans and compressors. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, *213*(4), 401-410. https://doi.org/10.1243/0954406991522356

Podobuiev, J. S., & Selezniev, K. P. (1957) *Theory and calculations of axial and radial compressors*. Maszgiz (in Russian).

Reunanen, A. (2001). Experimental and numerical analysis of different volutes in a centrifugal compressor [D.Sc. Thesis, Lappeemranta University of Technology].

Ris, W. F. (1951). *Radial compressors*. Maszgiz (in Russian).

Tuliszka, E. (1976). *Compressors, blowers and fans*. WNT (in Polish).

Van den Braembussche, R. A., & Hände B. M. (1990). Experimental and theoretical study of the swirling flow in centrifugal compressor volutes. *Journal of Turbomachinery*, *112*(1), 38-43. https://doi.org/10.1115/1.2927418

Walczak, J. (1999). *Radial compressors and fans. Theory, an investigations and optimization of compressor stage*. Publishing House of Poznan Society of Friends for Sciences (in Polish).

Wen, X., Mao, Y., Yang, X., & Qi, D. (2016). Design method for the volute profile of a squirrel cage fan with space limitation. *Journal of Turbomachinery*, *138*(8), Article 081001. https://doi.org/10.1115/1.4032537

Witkowski, A. (2004). *Rotary compressors. Theory, design, exploitation*. Publishing House of Silesian University of Technology (in Polish).

Yahya, S. M. (2012). *Turbines compressors and fans* (4th Ed.). Tata McGraw Hill Education Private Limited.

Metody Obliczeniowe Parametrów Spirali Stosowane w Projektowaniu Koncepcyjnym Sprężarek Promieniowych i Osiowo-Odśrodkowych

Streszczenie

W wielu projektach jednostopniowych sprężarek promieniowych i osiowo-odśrodkowych silników lotniczych turbośmigłowych i turbowałowych, odpowiednio uformowany kolektor umieszczony za bezłopatkowym lub łopatkowym dyfuzorem promieniowym, jest stosowany do zmniejszenia prędkości i zwiększenia ciśnienia statycznego strumienia powietrza. Dyfuzor spiralny jest jednym z głównych typów dyfuzorów. Spirala to kanał o zmiennym przekroju poprzecznym, który stopniowo rozszerza się w kierunku obrotu wirnika i obejmuje poprzedzający go dyfuzor z cylindrycznym otworem wlotowym. Jego parametry geometryczne powinny być odpowiednio dobrane, tak aby zapewnić prawidłową pracę spirali. W artykule przedstawiono dwie główne metody obliczania parametrów geometrycznych dyfuzora spiralnego: projektowanie swobodnego wiru (zasada stałego momentu pędu) i projektowanie według stałej średniej prędkości. Wymienione metody (zalecane do stosowania w projektowaniu koncepcyjnym sprężarki) opierają się na równaniu energii - równaniu energii przepływu ustalonego, równaniu ciągłości, pierwszej zasadzie termodynamiki, równaniu momentu pędu Eulera, funkcjach dynamiki gazów i definicjach stosowanych w teorii maszyn wirnikowych. Przeprowadzono szczegółową analizę parametrów geometrycznych różnych typów kolektorów. W artykule przedstawiono również przegląd wyników badań eksperymentalnych współczynnika strat ciśnienia całkowitego w spirali oraz zaproponowano metodę określania parametrów strumienia powietrza na wylocie spirali.

Słowa kluczowe: silnik turbinowy, sprężarka, kolektor