Improved methods of hydrodynamic characteristics for calculating and profiling input, transition and outlet pipes of the GTE transport

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Abstract. The present article deals with the results of the method development for calculating the flow in the channels of gas turbine tubes, basing on direct and inverse problems of hydrodynamics. An approach to optimize the shape of the channel, based on the boundary layer model is also discussed. Some results of experimental verification of the proposed approach for the transition pipe model are presented.

A certain improvement in the characteristics of the axial-flow compressor can be obtained by some more careful coordination of the blade and the compressor nozzles, considering them in the form of a system of elements: inlet pipe – axial compressor-transition pipe-centrifugal compressor-outlet pipe. The concept of "harmonization" of the compressor working elements should include not only the parameters on the calculated average radius or average of the surface current, but also the harmonization of fields of velocities in the sections between the flanges and the blade apparatus of the compressor. The speeds at the outlet of the inlet pipe determines the nature of the flow around the impeller blades of the first stage of the axial compressor, if they were designed to operate in a uniform flow. The same can be said about the operation of the input section of the impeller of the centrifugal compressor, which enters the flow after the transition pipe. The characteristics of the transition pipe depend significantly on the velocity field at the inlet, i.e. from the velocity field at the outlet of the axial compressor blade unit with its radial unevenness and high turbulence due to aerodynamic traces from the previous rows of blades.

Coordination and optimization of the system of these five elements is impossible without the presence of detailed characteristics of each of these elements under the specified conditions at the input of the degree of unevenness of the velocity field and the degree of turbulence of the flow, not to mention the known criteria M and Re. It should also be noted that in some cases, there are some power racks (or ribs in the channel of the branch pipe), aerodynamic traces of which give a step unevenness with additional radial unevenness at the ends of these racks. The reduction of the above mentioned step unevenness can be obtained by blowing a small amount of air into the area of the output edges of the power racks, and the radial unevenness in the area of the end sections can be affected by blowing air into the boundary layers on the surfaces of the housing and sleeve or by sucking these boundary layers.
Therefore, the task of creating a rational axial-flow compressor for the transport turbine engine with a given degree of total pressure increase should include the following stages:

1. Pre-distribution of the total degree of increase of the total pressure between the axial and centrifugal parts of the compressor, taking into account losses in the pipes.
2. Calculation and profiling of the inlet pipe for a given GTE layout.
3. Joint calculation of the inlet pipe with the first one or two stages of the axial compressor along the middle current line.
4. Calculation and profiling the transition pipe for a given GTE layout.
5. Joint calculating the transition pipe with the last one or two stages of the axial compressor along the middle current line.
6. Combining calculation transition branch pipe with the centrifugal stage.
7. Combining calculation of the outlet with a centrifugal stage
8. Profiling blade apparatus in a connecting section with the previous nozzles.
9. Refined distribution of the total degree of pressure increase between the axial and centrifugal parts of the compressor.

This paper considers the influence of losses in the pipes on the integral parameters of the compressor as well as a method of calculating the velocity fields in the output sections of the inlet and transition pipes under some specified conditions at the inlet to the pipe.

Transport gas turbine engines are usually located in some very cramped in size engine compartments. The air supply to the compressor is carried out through the inlet pipe, being of a rather complex structure, which leads to noticeable total pressure losses. Total pressure losses occur in the transition pipe between the axial and centrifugal parts of the compressor and in the outlet pipe. When we create a compressor, it is necessary to obtain a given degree of increase in total pressure, and it is advisable to allocate losses in the pipes as well as consider the integral characteristics of the compressor, taking into account both losses in the pipes and the characteristics of its stages.

The calculation is initially limited to a one-dimensional approach to the analysis of the gas flow in the pipe channel, considering the relationship between the coefficients characterizing the effect of the medium viscosity on the gas parameters and the gas flow through the channel of a given shape and size when there is no heat exchange with the environment through the wall.

As such coefficients we take:

1. Speed ratio \( \varphi = \frac{C}{C_S} \), equal to the ratio of the real velocity \( C \) to its value \( C_S \) in the flow without friction under the same conditions at the inlet to the channel.
2. Flow ratio \( \mu = \frac{\dot{G}}{\dot{G}_S} \), equal to the ratio of the actual mass flow rate \( \dot{G} \) to the flow rate \( \dot{G}_S \) in the same section of the channel without friction at the same values of static pressures \( (P = P_S) \) and total temperatures \( (T^* = T_S^*) \).
3. Total pressure conservation factor \( \sigma = -\frac{P^*}{P_S^*} \), equal to the ratio of the total pressure \( P^* \) in this section of the channel to the total pressure \( P_S^* \) in the same section without friction, when \( P_S^* = P_1^* \).

When creating the transport input and transition, the GTE nozzles are typically convergent, and outlet nozzle operates as a diffuser. When we have some data of the loss factors in the nozzles and characteristics of the compressor stages, it is possible to solve the problem of optimization of the flow part of the compressor as a whole, as well as optimize the system of the compressor pipe – stage.

When developing a new layout of the turbine engine, and if there are no results of experimental studies of the elements of the flow of turbo-machines, it is desirable to carry out at least an approximate design optimization of the blade with adjacent nozzles. The physical presence of the possibility of such optimization is due to the contradictory influence of the consumable gas velocity on the pressure losses in the nozzle and in the adjacent stages of the compressor. At a given value of the coefficient \( \xi \), losses in the nozzle grow with the growth of the reduced velocity \( \lambda \), and losses in the adjacent stage at a given value of the speed of the end of the working blade \( U_k \) have a minimum at
some optimal value of the flow coefficient \( \frac{C_a}{\omega_k} \) at a given pressure coefficient. Such operating conditions are usually implemented in gas blowers of nuclear plants and fans with a strong effect of losses in the pipes on the efficiency of the entire installation [3]. Also we are to take into account both the influence of radial clearance and elongation of the blades on the efficiency of a stage.

When profiling the pipes by the method of hydrodynamic characteristics, the solution of the inverse dynamics of the layout considerations provides a very wide diversity of channels pipes in the design of compressors transport GTE, although they correctly have an axisymmetric shape. It has already been noted above that the nozzle channel should provide gas supply to the flow part of the compressor itself with minimal losses for the system at a well-defined velocity field along the channel height for rational profiling of the blades along the radius.

When calculating the velocity field in an axisymmetric channel, two approaches are possible. The first approach solves the so-called inverse problem, when the channel contours are current lines from the system of singularities in the potential flow of an incompressible inviscid fluid. In this case, it is the distribution of pressure along the contour of the channel and the velocity field at the outlet of the channel in an analytical form that are convenient for subsequent calculation of the behavior of the boundary layer. Changing the system of features allows you to influence the distribution of pressure along the contour of the channel when changing the contour so as to obtain an improvement in the characteristics of the boundary layer. In this case, it is possible to set the problem of the optimal shape of the channel contour associated with this system of features.

The second approach solves the direct problem of determining the velocity field in the channel of a given shape. The solution is obtained in numerical form, while optimization is difficult.

Consider the solution of the inverse problem for several types of nozzles.

Let the layout considerations lead to the contour of the branch pipe shown in Figure 2. In the central fairing there is a thickening for the location of the bearing and auxiliary units, and the outer contour of the inlet pipe is connected to the contour of the fairing. A similar form of axisymmetric channel can be obtained by five hydrodynamic features: three vortex rings \( \Gamma_1, \Gamma_2, \Gamma_3 \) of the source \( Q_1 \) and the flow \( Q_2 \), located on the axis of symmetry, Figure 2. Vortex rings have the same direction of rotation and give one of the meridional current lines in the form of eights with zero flow rates between the rings.

The third point with zero speed is located on the axis of symmetry, where the velocities from each feature are directed along the axis, but have different directions (from the source \( Q_1 \) to the left, from the vortex rings \( \Gamma_1, \Gamma_2, \Gamma_3 \) and the flow \( Q_2 \) – to the right). Different abundance of the source and the drain allow to obtain an unclosed surface of the central fairing like a half-body, obtained by flowing around the source with a uniform flow.

Varying the values \( \Gamma_1, \Gamma_2, \Gamma_3, Q_1 \) and \( Q_2 \) and their relative positions (\( R_1, R_2, b, a, c, l \)), a very wide range of input device channel shapes can be obtained.
The procedure for determining the shape of the channel circuit is reduced to finding the values of the current function $\Psi_2$ at the points of a series of lines perpendicular to the axis of symmetry. As the first such line, it is convenient to take a straight MN at the outlet of the inlet pipe, and the points M and N correspond to the housing and sleeve at the inlet to the 1st stage of the axial compressor. Finding the distribution $\Psi$ along a series of lines parallel to the line MN, we can connect the points on these lines, in which $\Psi_{\Sigma_i} = \Psi_{\Sigma_M}$ and $\Psi_{\Sigma_j} = \Psi_{\Sigma_N}$.

To calculate the velocities at the points of the contour of the outer and inner boundaries of the branch pipe, it is necessary to use the value of the given flow rate $G$ and the distribution of the axial velocities along the MN line, expressed in terms of the value $\Gamma_1$, for example, see [4]. According to the found velocities, the pressure distribution is found, by which the behavior of the boundary layer and the friction losses in this branch pipe can be calculated.

As a result of this calculation, the contour of the branch pipe channel and the velocity field in its output section are determined simultaneously.

Then, the influence of compressibility and viscosity on the velocity field is taken into account by one of the known methods. It is possible to optimize the contour of the pipe from the point of view of frictional losses for the selected scheme abilities.

The contour of the transition pipe can also be obtained by the method of features, as which it is enough to take two vortex rings $\Gamma_1$ and $\Gamma_2$, Figure 3. Choosing the ratio between $R_1$, $R_2$ and C, it is possible to obtain the desired shape of the channel between the cross sections of the axial compressor outlet (1) and the centrifugal compressor inlet (2). The calculation of the pressure distribution along the transition pipe circuit is performed in the manner described above.

![Figure 2. Inlet pipe formed by 3 vortex rings $\Gamma_1$, $\Gamma_2$, $\Gamma_3$, the source $+Q_1$ and the flow $-Q_2$ different mobilities in the forward flow](image)

![Figure 3. The contour of the transition pipe formed by 2 vortex rings $\Gamma_1$, $\Gamma_2$](image)
To account for the twisting of the flow while maintaining its potentiality on the axis of symmetry it is a vortex cord that creates a circumferential projection of the velocity by the formula $r \cdot C_u = const$. If there is a flow twist, the pressure distribution along the channel contour will be different in accordance with the change in the absolute velocity $c = \sqrt{c_a^2 + c_u^2 + c_r^2}$ in the points of the contour.

The contour of the axisymmetric outlet pipe can also be obtained by vortex ring methods taking into account the twist according to the law of constant circulation. To obtain a practically appropriate circuit, a system of 3-4 vortex rings of different intensities located on a conic surface should be considered (Figure 4). It should be noted, however, that the use of the system of discrete vortex rings gives the "waviness" of the contour channel, which will impact mostly to do with the determination of the contour of the channel, and the calculation of pressure distribution in a smooth channel, to avoid waviness of the contour can be passed to a continuous distribution of vortex ring density at a conical surface, see, for example, the calculation of the ring of blades in an axially symmetric channel [4].

![Figure 4](image.png)

**Figure 4.** The contour of the axisymmetric branch pipe formed by vortex rings $\Gamma_1, \Gamma_2, \Gamma_3, \Gamma_4, \Gamma_5, \Gamma_6$, taking into account the twist according to the law of constant circulation

In [8], devoted to the study of the boundary layer and the resistance of a longitudinally streamlined free cylinder taking into account the influence of the transverse curvature of the surface, it was pointed out that in solving the problem of the turbulent boundary layer of axisymmetric bodies, the influence of the transverse curvature of the surface was either not taken into account at all, or partially. In most cases, it was assumed that the thickness of the boundary layer is to some little degree compared to the radius of curvature.

But neglecting the influence of transverse curvature leads to noticeable errors in the determination of local and total resistance coefficients. For the case of an external flow around a free cylinder, the effect of curvature leads to an increase in the local and total friction coefficients. Especially the effect of curvature affects the small numbers $Re_r = \frac{u_r r}{\nu}$ and the big ones $Re_x = \frac{u_x x}{\nu}$.

For axisymmetric channels, the effect of the transverse curvature of the surface is even less, since the ratio of the boundary layer thickness to the channel radius $\delta / r_{\omega}$ does not exceed one, and in our case it was less.

It should be remembered that the calculation of the axisymmetric boundary layer is somewhat more difficult to calculate than a flat boundary layer because the flow on the body of rotation, streamlined in the axial direction, also depends on the shape of the body due to the presence in the equation of inequality of the variable radius($x$) of the cross-section of the body perpendicular to the axis of rotation.

Let us compare the boundary layer equations for axisymmetric and plane tasks[6].
For a plane task

\[ \begin{align*}
\mathbf{I} \left\{ \mathbf{u} \cdot \frac{\partial \mathbf{u}}{\partial x} + \mathbf{\hat{v}} \cdot \frac{\partial \mathbf{u}}{\partial y} = \mathbf{u} \cdot \frac{\partial \mathbf{v}}{\partial x} + \mathbf{\hat{v}} \cdot \frac{\partial^2 \mathbf{u}}{\partial y^2} \\
\frac{\partial \mathbf{u}}{\partial x} + \frac{\partial \mathbf{v}}{\partial y} = 0
\end{align*} \]  

(1)

For the axisymmetric task in a curvilinear coordinate system, where: x is measured along the arc of the Meridian of the body of rotation, and y is normal to the wall

\[ \begin{align*}
\mathbf{II} \left\{ u \cdot \frac{\partial u}{\partial x} + v \cdot \frac{\partial u}{\partial y} = u \cdot \frac{\partial u}{\partial x} + v \cdot \frac{\partial^2 u}{\partial y^2} \\
\frac{\partial (ru)}{\partial x} + \frac{\partial (rv)}{\partial y} = 0
\end{align*} \]  

(2)

The value \( r \) in the second equation of the system (II) means the distance of the point of the body surface from the axis of symmetry.

Both systems differ from each other by the second equations, the continuity equation of the axisymmetric problem includes the radius \( r(x) \), which is absent from the continuity equation of the plane problem.

Work [6] indicates the existence of a relationship between axisymmetric and plane boundary layers, which allows to reduce the calculation of the axisymmetric boundary layer on the body of rotation to the calculation of the boundary layer for the plane problem.

The formulae of the Mangler transformation to convert the coordinates and velocities of the axisymmetric problem into the corresponding coordinates of the velocity of the plane problem have the following form:

\[ \begin{align*}
\bar{x} &= \frac{1}{L^2} \int_0^L r^2(x) dx; \\
\bar{y} &= \frac{r(x)}{L} \cdot y
\end{align*} \]  

(3)

\[ \begin{align*}
\bar{u} &= u; \\
\bar{\mathbf{v}} &= \frac{L}{r} \left( v + \frac{r'}{r} \cdot y \cdot u \right); \\
v &= \frac{L}{r} \cdot \left( \frac{\partial f}{\partial y} + \frac{L}{r} \cdot \frac{\partial f}{\partial r} \right)
\end{align*} \]  

(4)

where: L is the constant length. Bearing in mind the ratio:

\[ \frac{\partial f}{\partial x} = \frac{r^2}{L^2} \cdot \frac{\partial f}{\partial x} + \frac{r^2}{L^2} \cdot \frac{\partial f}{\partial y} + \frac{\partial f}{\partial r} \cdot \frac{r}{L} \]  

(5)

We make sure that the equations I after transformation (III) easily pass into equations (II).

Thus, the calculation of the boundary layer on the body of rotation can be reduced to the calculation of a flat boundary layer on a curved surface. After calculating the velocities \( u, v \) of the plane boundary layer, it is necessary to return to the velocities of \( u, v \) – axisymmetric boundary layer by means of transformation formulas (III).

Simplifying the calculations you can use the dependencies for a flat plate under gradientless flow. The results obtained should be considered approximate, since it does not take into account the spatial nature of the flow and the shape of the streamlined surface.

The method of sources and sinks was first applied by Rankin to the spatial problem of body flow. The idea of the method is to replace the body under consideration by a system of sources and drains located on the axis of rotation, so that the surface of the body serves as one of the surfaces of the current for the flow formed by this system of features.

It should be noted that the selection of the mentioned system of sources and effluents according to the predetermined shape of the streamlined body surface is very difficult. In work [5] a number of the simplest cases of definition of forms of surfaces which can be taken for borders of a streamlined body at a predetermined arrangement of sources is in detail considered.

But in our case, to determine the inner surface of the inlet pipe, the most suitable is a combination of the source and the flow of different abundances introduced into a uniform translational flow. Let us consider this case in a more detailed form (Figure 5).
The flow has two critical points K and L lying on the real axis, in which the velocity vanishes. For the point K \( V_z = 0 \) or, by entering the module \( z_k \), we’ll see

\[ V_z = V_\infty - \frac{Q_1}{4\pi} \frac{z_k}{z_k^2} + \frac{Q_2}{4\pi} \frac{(z_k+\beta)^3}{(z_k+b)^2} = V_\infty - \frac{Q_1}{4\pi} \frac{1}{z_k^2} + \frac{Q_2}{4\pi} \frac{1}{(z_k+b)^2} = 0 \]  

(6)

The exact value of the modulus \( z_k \) is determined from equation (1). The approximate value \( z_k' \) can be determined by the formula

\[ z_k' = \sqrt[4]{\frac{Q_1}{4\pi V_\infty}} \]  

(7)

An almost exact value \( z_k' \) can be determined by the method described below.

Similarly, reasoning, we define the module \( z_L \) for the point L.

\[ V_z = V_\infty + \frac{Q_1}{4\pi} \frac{1}{z_L^2} - \frac{Q_2}{4\pi} \frac{1}{(z_L-b)^2} \]  

(8)

The approximate value \( (z_L - b) \) is determined from equation

\[ (z_L - b) = \sqrt[4]{\frac{Q_2}{4\pi V_\infty}} \]  

(9)

Then, by taking over three meaning \( z_L = 0,0\sqrt[4]{z_L}, z_L'and 1,1z_L' \) determine \( V_{zL} \) from the equation (3), then build a graph \( V_{zL} = f(z_L) \) (see fig. 6), get the exact value of \( z_L \). The exact value is determined similarly \( z_K \).

The equation of the contour of the half-body is determined under the condition \( \Psi = 0 \) [5].

\[ \Psi = -\frac{1}{2} \frac{x^2}{y^2} \cdot V_\infty - \frac{Q_1}{4\pi} \left[ 1 - \frac{z}{\sqrt{x^2+y^2}} \right] + \frac{Q_2}{4\pi} \left[ 1 - \frac{x-b}{\sqrt{(x-b)^2+y^2}} \right] \]  

(10)

For the accepted value \( z_l \) count the values \( \Psi_l \) for the series of values \( \tau_j \) and build a graph \( \Psi_l = f_1(\tau_j, z_l) \) see Figure 6. Connecting the points with \( \Psi_l = 0 \), we obtain the desired contour of the body placed in the translational flow.

The idea of the proposed method of profiling the inlet of the annular curved channels lies in the combination of the method of source – drains and method of vortex rings. The incoming flow superimposed on the source and flow of different mobilities is created by a system of vortex rings. The inner surface of the pipe, having the form of a half-solid, is a zero current line and is determined basing on the zero value of the current function of the system of all features.
For Fig. 7 the proposed layout of the features for the expected contour of the inlet pipe is presented. The equation of the current line for this scheme is as follows

$$\psi = -\frac{r_1 R_1}{2\pi} \cdot \psi'(\bar{r}_1; \bar{z}_1) - \frac{r_2 R_2}{2\pi} \cdot \psi'(\bar{r}_2; \bar{z}_2) - \frac{r_3 R_3}{2\pi} \cdot \psi'(\bar{r}_3; \bar{z}_3) - \frac{Q_1}{4n} \cdot \left[ 1 - \frac{x}{\sqrt{x^2 + r^2}} \right] + \frac{Q_2}{4n} \cdot \left[ 1 - \frac{x-b}{\sqrt{(x-b)^2 + r^2}} \right]$$

(11)

Then define, as mentioned above, the values \(\Psi_{\Sigma}^*\) for a series of values \(r_j\) in each section \(z_i\) by equation (5) and build the graph of the function \(\Psi_{\Sigma}^* = f(r_j, z_i)\), see fig. 5. Connecting the points with \(\Psi_{\Sigma}^* = 0\), we obtain the contour of the central body of the inlet pipe.

The contour of the central body must pass through point B (see Figure 7) with a given value of the diameter of the sleeve \(d_{w1}\) at the inlet to the flow part of the axial compressor. If this does not happen immediately, then the scale of the current lines picture is changed so that the point B falls at a given distance from the compressor axis \(d_{w1}/2\). Then the position of point A is determined by the given value \(d_k\) and the value of the current function \(\Psi_{\Sigma}^*\) on the line 1-1 input to the axial compressor. \(d_{w1}\) and \(d_k\) – are determined.

$$\bar{r}_1 = \frac{r}{R_1}; \bar{r}_2 = \frac{r}{R_2}; \bar{r}_3 = \frac{r}{R_3}; \bar{z}_1 = \frac{z_1}{R_1}; \bar{z}_2 = \frac{z_2}{R_2}; \bar{z}_3 = \frac{z_3}{R_3}$$

(12)
Defining a value $Ψ_{ΣA}^*$, for a given value $d_0$ (point A), finding in each section $z_i$ the coordinates of $r_j$ for the same values $Ψ_{ΣA}^* = Ψ_{ΣA}^*$ and connecting them we get the contours of the outer surface of the channel.

The value of the current function included in equation (5) is represented in dimensionless form with a constant multiplier $\frac{r_1}{2\pi}$, where the value $I_1$ is unknown. To determine the value of $I_1$ determine the speed value $V^*$ in section 1-1, Figure 8.

![Figure 8. Scheme for determining the value of $I_1$ in section 1-I](image)

The speed at section 1-1 is determined as the algebraic sum of the speeds of all the features.

$$V^*_{Σ} = V^*_{z_1} + V^*_{z_2} + V^*_{z_3} + V^*_{Q_1} + V^*_{Q_2}$$  \hspace{1cm} (13)

Express values $Γ_2$, $Γ_3$, $Q_1$, $Q_2$ through $Γ_1$

$$V^*_{Σ} = \frac{Γ_1}{2πR_1}, \left[ V^*_{z_1} + \frac{Γ_2}{2πR_2} \cdot V^*_{z_2} + \frac{Γ_3}{2πR_3} \cdot V^*_{z_3} + \frac{Q_1}{2π} \cdot \frac{z_{10}}{r_1} - \frac{Q_2}{2π} \cdot \frac{z_{20}}{r_2} \right]$$  \hspace{1cm} (14)

Substituting the value $z_1$, $z_2$, $z_3$, $z$, $Z_{10}$, $Z_{20}$, $R_1$, $R_2$, $R_3$, $Γ_1$, $Γ_2$, $Γ_3$, $Q_1$, $Q_2$ in equation (13) we define the total dimensionless velocity $V^*_{Σ}$ from 5 features system for several values of $Γ_1$.

Then setting three values for $Γ_1$ and determining $ε(λ, k)$ for several values $r_i$ we find the value of the dimensionless parameter $ε(λ, k) \cdot V^*_{Σ} \cdot r$ and after graphical integration we determine the flow rate $G$ by the equation (16):

$$G = r^2_0 \cdot \frac{Γ_1}{R_1} \cdot ρ^* \cdot \int_{r}^{1} ε(λ, k) \cdot V^*_{Σ} \cdot r \cdot d r$$  \hspace{1cm} (16)

Then building a graph of dependence $G = f(Γ_1)$ (see Figure 9) and putting aside the required air flow, for example, equal to 4.3 kg/s, determine the value $Γ_1$.

After determining $Γ_1$ find the velocity distribution $V = \sqrt{V^2_{Σ} + V^2}$ and $λ$ numbers on the channel surface, and then the pressure distribution from the equation

$$\frac{P}{ρ^*} = \left(1 - \frac{k-1}{k+1} \cdot λ^2\right)^{\frac{k}{k-1}} = π(λ, k)$$  \hspace{1cm} (17)

or

$$P = P^* \cdot \left(1 - \frac{k-1}{k+1} \cdot λ^2\right)^{\frac{k}{k-1}} = P^* \cdot π(λ, k)$$  \hspace{1cm} (18)
\[ \pi(\lambda, k) \text{ determining with } k=1.4 \text{ and } \lambda = \frac{V}{a_{eq}}, \text{ where all braking parameters are taken to be equal to the ambient air at } P^*=760\cdot133.2 \text{ MPa and } T^*=288.3 \text{ K.} \]

**Figure 9.** The graph of the G = f (Г1) dependence

Similar reasoning is used when profiling the reducer, the method of vortex rings.

A vortex ring is a feature in an axisymmetric flow just like a dipole, source, or vortex in a flat flow.

By placing the dipole in a plane-parallel flow, we obtain a flow around a circular cylinder. The selected feature gave us a current line in the form of an almost interesting circuit in the plane of the liquid flow.

Reasoning this, it is possible to receive an axisymmetric channel, accepting two surfaces of a current of a vortex ring for channel walls.

Using a number of vortex rings, it is possible to obtain axisymmetric channels of various shapes, although with the growth of the number of rings, the computational difficulties of this method grow.

The method is the opposite, since the shape of the walls is obtained in the course of solving the problem.

For Fig. 10 the proposed scheme of location of features for a given circuit of the transition pipe is presented.

Write the equation of the current line (see [4]).

\[ \Psi = -\frac{\Gamma_1 R_1}{2\pi} \cdot \Psi^* (\frac{r_1}{R_1}; \frac{z_1}{R_1}) - \frac{\Gamma_2 R_2}{2\pi} \cdot \Psi^* (\frac{r_2}{R_2}; \frac{z_2}{R_2}) = -\frac{\Gamma_1 R_1}{2\pi} \cdot \left[ \Psi^* (\frac{r_1}{R_1}; \frac{z_1}{R_1}) + \frac{\Gamma_2}{\Gamma_1} \cdot \frac{r_2}{R_2} \cdot \Psi^* (\frac{r_2}{R_2}; \frac{z_2}{R_2}) \right] \] (19)

Then determine the value \( \Psi_1 \) for the series of values \( r_i \) in every section \( z_i \) by equation (19) and build a graph \( \Psi_1 = f (r_i, z_i) \)

Connecting points with equal values \( \Psi_1 \) we obtain the contours of the inner and outer surfaces of the transition pipe.

By defining the \( \Psi \) values for the given values \( d_k \) and \( d_{en} \) at the entrance to the transition pipe between the axial and centrifugal parts of the compressor, we define in each section \( z_i \) coordinates \( r_i \) for the same values \( \Psi \), equal at the inlet to the pipe, and connecting them, we obtain contours of the inner and outer surfaces of the channel.

\[ r_1 = \frac{r}{R_1}; \quad r_2 = \frac{r}{R_2}; \]
\[ z_1 = \frac{z}{R_1}; \quad z_2 = \frac{z}{R_2}; \]

The values of the current function obtained from equation (19) are presented in dimensionless form and have a constant multiplier \( \frac{\Gamma_1 R_1}{2\pi} \) where the value \( \Gamma_1 \) is unknown.
For determining the value \( \Gamma' \) determine the speed value \( v_z \) in section 1-1.

\[ V_z = V_{z1} + V_{z2} \]  \hspace{1cm} (20)

Express the value \( \Gamma' \) through \( \Gamma \)

\[ V_z = \frac{\Gamma_1}{2\pi R_1} \cdot V_{z1}^* (r_1; z_1) + \frac{\Gamma_2}{2\pi R_2} \cdot V_{z2}^* (r_2; z_2) \]  \hspace{1cm} (21)

Substituting the value \( z_1, z_2, R_1, R_2 \) in equation (22), take out a constant multiplier \( \frac{\Gamma_1}{2\pi R_1} \), and then determine the total dimensionless velocity \( V_z^* \) from a system of features for several values \( \Gamma_1 \).

Next, the method of profiling the transition pipe is similar to the method described earlier for the inlet pipe.

1. Given in this work, the materials indicate the need for a thorough accounting of the losses in the pipes of the compressor, as well as to the need for careful coordination of the design of the blade of the first compressor stage with a velocity field at the outlet of the previous nozzle.

2. There is a physical possibility of optimization of the system of the branch pipe-compressor stage, for which it is necessary to build mathematical models of both the branch pipe and the stage of the axial and centrifugal compressor.

3. The proposed system of hydrodynamic features makes it possible to obtain the contours of the input transition and output pipes that meet the layout requirements of the transport GTE [9].

4. Further improvement of the characteristics of the nozzles can be obtained with a certain change in the radius of the vortex rings, which will lead to a change in the contours of the outer and inner contours of the nozzles at the same air flow rate in the inlet section.

5. Comparison of the obtained data on the distribution of static pressure with the experimental data for the transition pipe given in [7] showed that for the external and internal contours of the channel close in shape, a similar nature of the change is observed, which suggests the correctness of the chosen calculation method.

6. It is necessary to carry out a detailed calculation of the boundary layer characteristics on the inner and outer contours of the branch pipe, taking into account its transverse and longitudinal curvature, although the estimates already allow to start calculating the profiling of the centrifugal compressor blade apparatus.
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