4

Optimization of the Axial Turbine Parameters Along the Stage Radius

4.1 Formulation of the Problem

Mathematical models of gas and steam turbines stages, discussed above, allow to put the task of their geometry and gas-dynamic parameters optimization. This optimization problem is solved by the direct problem of stage calculation. The reason for this are the following considerations:

- it is most naturally in optimizing to vary the geometry of the blades;
- in the streamlines form refinement it is convenient to use well-established methods for the solution of the direct problem in the general axisymmetric formulation;
- only a direct problem statement allows to optimize the stage, taking into account the off-design operation;
- For the stages to be optimized, assumed to be given:
- the distribution of the flow at the stage entrance;
- the form of the meridian contours;
- the number of revolutions of the rotor;
- mass flow of the working fluid;
- averaged integral heat drop.

In general, you want to determine the distribution along the certain axial sections of angles α_1 and β_2 to ensure maximum peripheral efficiency of the stage:

$$\eta_u = \int_0^{\psi^*} L_u d\psi \Big/ \int_0^{\psi^*} h_0 d\psi \,, \tag{4.1}$$

Where $L_u = u_1 c_{1u} - u_2 c_{2u}$; $\quad h_0 = L_u + \Delta h_s + \Delta h_r + c_2^2/2$.

Restriction on the heat drop, that determines the back pressure, is given in the form:

$$\int_0^{\psi^*} h_0 d\psi = h_{0m}\psi^* . \tag{4.2}$$

The effective angles of cascades may be restricted:

$$\alpha_{1\max} \geq \alpha_1 \geq \alpha_{1\min}; \quad \beta_{2\max} \geq \beta_2 \geq \beta_{2\min} . \tag{4.3}$$

Here the inlet geometric angle of the rotor we assume equal to the angle of the inlet flow. Selection of the optimal angle β_{1g} can be achieved solving an optimal profiling problem.

The objective function (4.1) can be calculated with the known distribution of the kinematic parameters of the flow in the gaps, which are determined by solving the direct problem of the stage calculation using the models set out in Chapter 2.

Mathematically, the assigned stage optimization problem has been reduced to a problem of the theory of optimal control of distributed parameter systems, including integrated criterion of quality (4.1) and system of constraints, which comprises:

- a system of equations in section 1 for the nozzle (2.39);
- a system of equations in section 2 of the rotor (2.40);
- isoperimetric condition (4.2), ensuring operation at a given stage heat drop;
- restrictions on the control variables (4.3).

The velocities c_1 and w_2, and the radii r_1 and r_2 are the phase variables; the independent variable stream function ψ plays the role of time.

From physical considerations it is obvious that the control functions $\alpha_1(\psi)$ and $\beta_2(\psi)$ must be sufficiently smooth, at least continuously differentiable, i.e. does not have discontinuities of the first kind, and kinks. For this purpose it is convenient to use parametric angles as dependencies

$$r_1^{m_1} \operatorname{ctg} \alpha_1 = \text{const}; \quad r_2^{m_2} \operatorname{ctg} \beta_2 = \text{const}, \tag{4.4}$$

which allow to investigate the influence of coefficients m_1 and m_2 on the stage efficiency.

The parameters m_1 and m_2 characterize twist angles gradients. For $m > 0$ are obtained the angles increasing to the periphery (direct twist) and for $m < 0$ – decreasing (reverse twist). Twist law $c_u r = \text{const}$ corresponds to the values $m_1 = 1$, $m_2 = -1$, that under the simplified equation of radial equilibrium, provides a minimum of the output velocity losses for the stage with cylindrical contours.

4.2 The Impact of Leaks on the Axial Turbine Stages Crowns Twist Laws

Significant impact on the stage efficiency have leakage of the working fluid through the seal gaps and discharge openings. The dependence of the leakage (and associated losses) of the stage bounding surfaces parameters can dramatically affect the distribution of the optimal parameters along the radii and, hence, the spatial structure of the flow therein. The latter, in turn, is determined by the shape and twist law of guide vane and impeller.

Development of algorithms for the axial turbine stages crowns twist laws optimization demanded the establishment of appropriate in the terms of computer time methods for calculating the quantities of leaks and losses on them, allowing the joint implementation of the procedure for calculating the spatial parameters of the flow in the stage.

The leakage calculation is necessary to conduct together with a spatial calculation step, as the results of which the parameters in the calculation sections are determined, including the meridian boundaries of the flow path. The flow capacity depends on the clearance (or leakages) values, in connection with which main stream flow calculation is made with the mass flow amplification at fixed the initial parameters and counter-pressure on the mean radius, or clarifying counter-pressure at fixed initial parameters and mass flow. The need for multiple stage spatial parameters calculation (in the optimization problem the number of direct spatial calculations increases many times) demanded a less time-consuming, but well reflecting the true picture of the flow, methods of spatial stage calculation in the gaps described above (Fig. 2.3).

When calculating stage in view of leakage the continuity equation is convenient to take as [8]:

$$\frac{\partial \psi}{\partial r} = \mu r \rho w_s \cos\theta , \tag{4.5}$$

where μ – the mass transfer coefficient, which allows to take into account changes in the amount of fluid passing through the crowns, and at the same time to solve a system of ordinary differential equations in sections in front of and behind the impeller like with a constant flow rate.

The leakage mass transfer coefficients [13] is defined as follows

$$\mu_{1leak} = \frac{G_0}{G_1} = 1 + \frac{G_d}{G_1} ;$$

$$\mu_{2leak} = \frac{G_0}{G_2} = 1 + \frac{G_d}{G_2} + \frac{G_r}{G_2} - \frac{G_h}{G_2} .$$

In the case of wet steam flow with loss of moisture, crown overall mass transfer coefficient is given by

$$\mu_i = \mu_{i,\,leak} \psi_{m,\,i}, \quad (i = 1, 2),$$

where $\psi_{m,\,i}$ – flow coefficient, is usually determined in function of the degree of humidity and pressure ratio [8].

As shown in Chapter 2, the calculation of stage spatial flow with the known in some approximation the shape of the streamlines is reduced to the solution in the sections $z_1 = \text{const}$ and $z_2 = \text{const}$ (Fig. 2.3) a system of ordinary differential equations (2.39) and (2.40), wherein as the independent.

As discussed in Chapter 2, the solution of (2.54) (2.55) for a given flow rate is reduced to finding the roots of the two independent transcendental equations in the hub velocities c_{1h}, w_{2h}.

For a given backpressure to the number of defined values the parameter ψ^* is added and the problem reduces to solving a system of three equations. As a third equation the heat drop constraint (2.45) is added which can be symbolically written as

$$h\left(c_{1h}, w_{2h}, \psi^*\right) - h_0 = 0. \tag{4.6}$$

Systems of equations are solved using the methods of nonlinear programming.

The calculation of stages with reverse twist using the proposed method allows to obtain a valid reaction degree gradient and circumferential velocity component of the stage, while the calculation provided cylindricality flow gives results that differ significantly from the experimental data. The technique allows to take into account also the effect of the rotor twist law on the parameters distribution in the gap after stator. This is evidenced by comparison of stages with the same nozzle assembly.

Effect of D_m/l ratio

With the methods described above, you can put the task of the stage parameters optimization along the height, taking into account the spatial flow and leakages. To explain the physical meaning of certain optimum blade units twist laws depending on different characteristics of stages, such as D/l ratio, radial gap, level of reaction degree at the mean radius, the presence of suction, and other factors, there is convenient to use the angle dependencies in the form (4.4).

The stages characteristics changing may be presented as constant level lines (isolines or topograms) in the plane of the variables m_1 and m_2. Computational studies were subjected to axial turbine stages with $D_m/l = 19\ldots3.2$, which have been tested on an experimental air turbine [13]. Some of them are at the same axial dimension and D_m/l have different levels of reaction at the mean radius that allows us to estimate the impact of the last factor on the crowns optimal twist laws. Separately the impact of radial clearance and suction at the root on optimal stage parameters was studied.

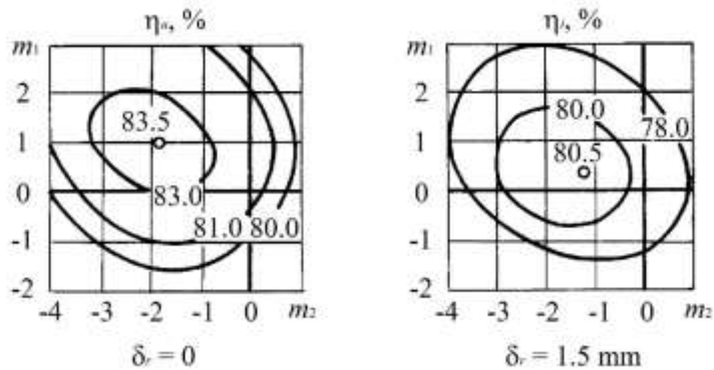
Influence of leakage through the radial gap

An important part of the kinetic energy loss in the axial turbine stage is a loss from radial clearance leakage, which is defined on the one hand, design and dimensions of the peripheral seal of the rotor blade and on the other – the pressure difference in the axial clearance in the outer radius. The analysis shows that in the stages of steam turbines for the amount of leakage significantly affects the D_m/l ratio: in the stages with relatively short blades, where δ_r/l_b is large, leakage losses greater of stages with a small D_m/l ratio, despite the higher degree of reaction at the outer radius of the latter. Higher losses from leaks have stage without rim seals.

Large turbine units operating experience shows, that the radial clearances may increase from 1.5 mm to 5 mm, which results in reduced efficiency due to leaks more than 2 %. In some turbines due to increased near-the-shroud gaps efficiency drops 2…3 % or even 5 %.

In order to analyze the impact of the radial clearance leakage losses to optimal axial turbine stage crowns twist laws calculation were conducted for the above stages with all sorts of combinations of parameters m_1 and m_2 and the radial clearance values. As a result of numerical experiments parameters level lines built that characterize the efficiency in the plane of m_1, m_2.

As an example, the results of the calculations are shown in Fig. 4.1, 4.2. Each point of the topogram was produced by the method of the spatial calculation in gaps with the streamlines refinement. The calculations were performed in different statements: with a given flow rate with a predetermined heat drop, with a predetermined flow rate and heat drop adjusting when you change the angle α_{1m}.



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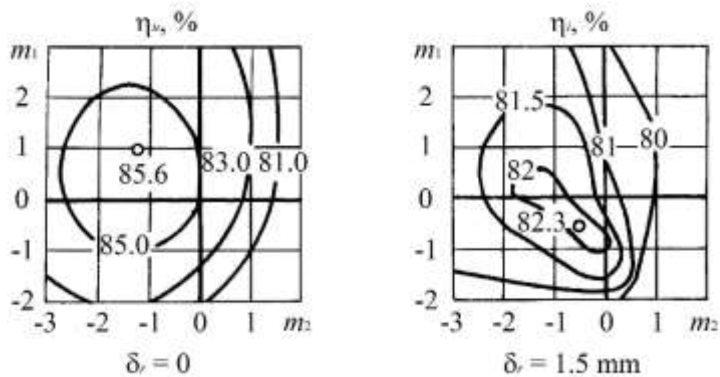

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Figure 4.1 *Dependencies of stages I and II efficiency with a different relative cascades width values with the* $D_m/l = 3.6$ *on the blades twist laws at different radial clearances.*

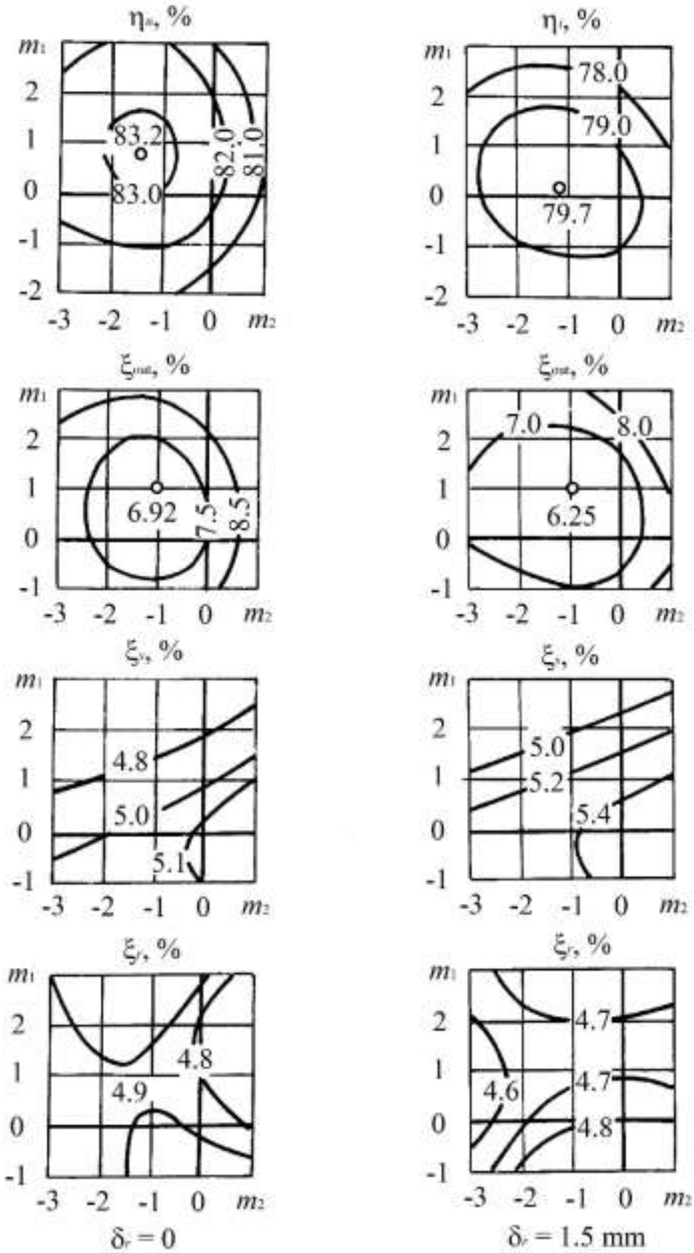

Figure 4.2 *Dependence of the relative losses in stage II with* $D_m/l = 3.6$ *on the blades twist laws at different radial clearances.*

Effect of parameters m_1, m_2 on the reaction gradient degree varies with different diameter to blade length ratios. Thus, for stages with relatively long blades ($D_m/l < 5$) substantial reaction gradient alignment has been observed when $m_1 = -1$, whereas for short blades ($D_m/l > 10$), requires values m_1 reduction up to –8…–10. Significant impact on the reaction gradient degree with the reverse twist of the guide blades have also stage axial dimensions, especially nozzle relative width (or chord). At lower values the effect of the reaction gradient degree leveling is stronger. Reaction gradient degree alignment not only affects the magnitude of flow leakage and loss of these, but also manifested in the increase of uneven circumferential velocity component along the radius for the rotor, which leads to increased losses from the exit velocity the more (depending on m_1 and m_2) the smaller the D_m/l ratio. The value of the leakage losses in the radial gap determined by the relative leakage flow rate, which depends, inter alia, on the radial clearance size and its design.

The results of numerous calculations indicate that the crowns optimum twist laws (parameters m_1 and m_2) for the stages of the various D_m/l ratio at different values of radial clearance is mainly determined by the ratio between the amount of output velocity losses, and losses from leaks in the radial clearance. Influence of hydraulic losses in the guide vane and the rotor has a significant impact on the level of the degree of reaction for very small leak quantities into over-shroud space.

At zero clearance maximum of peripheral efficiency of the cylindrical stage for small D_m/l located in a neighborhood of the point with a minimum exit loss (Fig. 4.1, 4.2): the guide vanes twist close to the $c_u r = \text{const}$ law, and impeller should be twisted a little more intense ($m_2 = -1 \ldots -2$).

With increasing D_m/l ratio the maximum peripheral efficiency shifts toward twist laws with $m_1 > 1$ due to the hydraulic losses influence. The amount of displacement depends on the method of calculation (with a given flow rate or heat drop) and the reaction degree at the mean radius: offset is stronger when calculating with a predetermined flow rate due to changes in the angle α_1 at the middle (on flow rate) radius as a result of the streamlines lifting, as well as an increase in the average reaction degree.

The increase of the relative radial clearance leads to a shift of the maximum internal stage efficiency point in the direction of m_1 downward and m_2 increase, which is a sharper, then D_m/l relation is greater. This entails the reaction degree gradient alignment due to the streamlines inclination to the hub after the stator and some thrown at the periphery of the impeller. For example, in sages with $D_m/l = 8.3$ in the absence of leakage in a radial gap some flow preload to the periphery in the gap between the vanes is expedient. As the radial clearance increases it is appropriate to decrease the level of the reaction degree at the mean radius and its gradient. When $\delta_r = 1.5$ mm is advantageous to almost completely eliminated the reaction gradient ($m_1 = -4 \ldots -5$, $m_2 = 0 \ldots 1$). The calculated results are in good agreement with the experimental study [13].

In the stages with even shorter blades ($D_m/l = 19$) for large radial clearances m_1 optimum value drops to –9…–11, and m_2 increases to 4…5. For large D_m/l values the stages with reverse twist for all real values of clearance have higher efficiency than the stages of traditional design. Winning increases with decreasing gap and the degree of reaction at the mean radius. Experimental research of stages $D_m/l = 19$ [13] fully confirms the conclusions.

The character of the stage relative loss change qualitatively is the same for all types of stages, with both small and large D_m/l ratios (Fig. 4.2). Isolines of the exit velocity loss form closed curves, surrounding the minimum point with $m_1 = 1$, $m_2 = -1$, corresponding to the constant circulation twist law. Isolines of the relative hydraulic loss regardless of the size of radial clearance are in the nature of saddle points: the relative losses in the guide vanes have in the saddle point m_1 maximum and m_2 minimum, while the relative losses in the rotor blades on the contrary, have m_1 minimum and m_2 maximum. With the D_m/l ratio increase hydraulic losses in the crowns becoming less dependent on the laws of another crown twist, acquiring the form of lines extended along the respective axes. This applies in particular to losses in the guide vane.

The leakage losses in the radial clearance in the plane of the variables m_1, m_2 achieved the highest value in the upper left corner of the topogram, where the peripheral degree of reaction is maximal, and the least – in the lower right corner, where the reaction degree gradient is minimal.

Influence of suction in the near-the-hub gap

To investigate the suction effect on the crowns best twist laws at fixed

parameters on the mean radius were selected three experimental air turbine stages, with different blades elongation and reaction degree at the mean radius ($D_m/l = 3.6, 8.3, 14.1$ and $R_m = 0.2, 0.02, 0.01$, respectively). Calculations were made for various values of the radial gap and flow suction introduced by changing the reduced gap of the diaphragm seal, to provide thereby suction value in stages with $D_m/l = 3.6$ – 0.5 %, with $D_m/l = 8.3$ – 1 % and with $D_m/l = 14.1 - 2$ %.

The influence of the flow suction does not change the conclusions regarding the optimal crown twist laws made above. It should be borne in mind that in actual turbine stages discharge holes presence results in a large impact on the suction flow of the pressure difference at the inner radius of the impeller. When properly selected reaction degree in the root and the appropriate size of the discharge holes relative flow rate of the jet can be almost reduced to zero.

4.3 The Axial Turbine Stage Optimization Along the Radius in View of Leakages

The above numerical study results, confirmed experimentally, show, that leakages significantly affect the axial turbine stage crowns optimal twist laws. With a decrease in the length of the rotor blade (increase of D_m/l ratio) this effect is amplified.

In this regard, the problem arises of determining the guide vanes and rotor optimal twist laws for a given stage geometry, inlet parameters, the rotor angular velocity, flow rate and heat drop. We restrict ourselves to the task of practically important case of the blades angles specification in the form (4.4). At the same time, while setting the flow and heat drop together, thermal calculation is performed by adjusting one of the angles α_{1m} or β_{2m}. Described below optimization technique based on repeated conduct this kind of thermal calculations for the purpose of calculating the internal stage efficiency depending on one of α_{1m}, β_{2m} angles, and the exponents m_1, m_2 in the expression (4.4).

Assume that the control variables are β_{2m}, m_1 and m_2, whereby the back pressure at a predetermined flow rate must be specified by changing the angle

α_1 at the mean radius. The problem of the thermal stage calculation is written as

$$r_1' = f_{11}(\alpha_{1m}, r_1, c_1); \quad c_1' = f_{12}(\alpha_{1m}, r_1, c_1);$$
$$r_1(0) = r_{1m}; \quad r_1(\psi^*) = r_{1t};$$
$$r_2' = f_{21}(\alpha_{1m}, r_2, w_2); \quad w_2' = f_{22}(\alpha_{1m}, r_2, w_2);$$
$$r_2(0) = r_{2h}; \quad r_2(\psi^*) = r_{2t}; \quad h(c_{1h}, w_{2h}, \alpha_{1m}) = 0,$$

and its numerical solution is based on finding the roots of transcendental equations

$$\left.\begin{array}{l} \tilde{r}_{1t}(c_{1h}, \alpha_{1m}) = r_{1t}; \\ \tilde{r}_{2t}(c_{1h}, w_{2h}, \alpha_{1m}) = r_{2t}; \\ h(c_{1h}, w_{2h}, \alpha_{1m}) = 0. \end{array}\right\} \tag{4.7}$$

After the solution of (4.7), which is conducted with the specification form of the stream lines, leakage values, velocity and flow rate coefficients, internal stage efficiency calculated as a function of three variables β_{2m}, m_1, m_2.

Thus, the stage optimal design problem with a maximum internal efficiency reduced to a nonlinear programming problem:

find $$\max_{\beta_{2m}, m_1, m_2} \eta_i = \frac{N}{h_0}, \tag{4.8}$$

where h_0 is calculated by the formula [13]:

$$h_0 = N + \Delta h_s + \Delta h_r + \Delta h_{out} + \Delta h_d G_d + \Delta h_r |G_r| + \Delta h_{bh} |G_{bh}| +$$
$$+ \left[\Delta h_h + \frac{1}{2}(c_h \sin \gamma_h)^2\right] |G_h| + \Delta h_{mix}, \tag{4.9}$$

Unconditional maximization (4.8) does not meet the fundamental difficulties. Physically seems justifiable division of the optimization problem (4.8) on two related subtasks:

- determination of optimal angles at the mean radius under the certain blades twist laws;
- selection of optimal parameters m_1 and m_2 at specified angles.

Thus, the general problem (4.8) can be solved iteratively by alternately solving subtasks:

$$\max \eta_i = \frac{N}{h_0} \quad \text{at} \quad m_1, m_2 = \text{const}; \tag{4.10}$$

$$\max_{m_1,\, m_2} \eta_i = \frac{N}{h_0} \quad \text{at} \quad \beta_{2m} = \text{const}; \tag{4.11}$$

This approach is analogous to the component-wise optimization, which is a special case of the known method of the coordinate descent (Gauss-Seidel). The first of the sub-tasks (4.10) is solved by searching the extremes of functions of one variable. The second sub-task (4.11) can be solved by direct search of two variables function extremum. The combination of a one-dimensional search of the best angles at the mean radius and a direct search of optimum parameters m_1, m_2 was the most reliable way of the problem (4.8) numerical solution, providing the finding of the global maximum of the objective function even in the presence of local extrema in the topogram plane.

As a practical application of the developed technique of the turbine stage spatial optimization in view of leaks were upgraded cylinders of high and intermediate pressure of the steam turbine with 200 MW capacity. Relation D_m/l varies from 25 (II stage of HPC) to 4.8 (VII last stage of IPC) (control stage was not considered), the number of stages in the HPC and IPC was 6 and

7, respectively. As initial were taken the stages with the manufacture's blade angles at the mean radius. If $D_m/l > 10$, rotor blades assumed cylindrical, and if $D_m/l < 10$ – twisted by constant circulation law. Modernization was carried out at regular radial gaps ($\delta_r = 0.001D_t$).

The data allowed to build dependence of the parameters m_1 and m_2, characterizing crowns twist laws on the ratio D_m/l (Fig. 4.3).

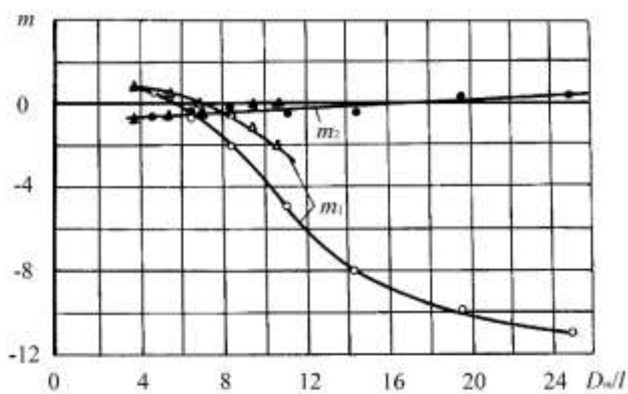

Figure 4.3 *Dependencies of the parameters m_1 and m_2, defining the crowns twist laws of powerful steam turbine HPC and IPC stages on the D_m/l ratio.*

Comparison of the effectiveness of IPC sections, consisting of original and optimized stages showed that efficiency of the last 0.65% above baseline. With increased radial clearances the gain increases.

The possibilities of this optimization method can also be illustrated by modern powerful (500 MW) steam turbine IPC upgrading example. 500 MW turbine intermediate pressure cylinder consists of 11 stages in the range $D_m/l = 10.8 \ldots 3.6$.

Optimization calculations have shown the expediency of the crowns twist with m_1 and m_2, the values and the magnitude of which change depending on

D_m/l are denoted with triangles in Fig. 4.3. Reducing exponents m_1 for stages with the same D_m/l values from the turbine 500 MW compared to 200 MW turbine can be explained narrower guide vanes (lower B_t/l values) in the IPC of the first turbine. Carried out optimization calculations have shown 0.45% improved cylinder efficiency.

4.4 The Effect of Tangential Lean on the Characteristics of Axial Turbine Stage

One means of the flow control in the axial turbine stage is the use of blades with non-radial setting. In this case, there is a non-zero the blade surface lean angle.

Vortex equation for the case of flow in a rotating crown can be written as:

- projected on the radial direction:

$$\left(1+\operatorname{ctg}^2\beta\right)w_s\frac{\partial w_s}{\partial r}-w_s^2\left(x\cos\theta+\frac{\partial\ln w_s}{\partial s}\sin\theta-\frac{\operatorname{ctg}^2\beta}{r}-\frac{1}{2}\frac{\partial\operatorname{ctg}^2\beta}{\partial r}\right)+$$
$$+2\varpi w_s\operatorname{ctg}\beta=\frac{\partial H}{\partial r}-T\frac{\partial s}{\partial r}-F_r-f_r\,; \tag{4.12}$$

- in the projection on the circumferential direction:

$$F_u=w_s^2\left(\frac{\sin\theta\operatorname{ctg}\beta}{r}+\frac{\partial\operatorname{ctg}\beta}{\partial s}+\frac{\partial\ln w_s}{\partial s}\operatorname{ctg}\beta\right)+2\varpi w_s\sin\theta-f_u\,, \tag{4.13}$$

where
$$\operatorname{ctg}\beta=\operatorname{ctg}\beta_p\cos\theta+\operatorname{tg}\delta\sin\theta\,; \tag{4.14}$$

$$F_r=-\operatorname{tg}\delta\,F_u\,. \tag{4.15}$$

Substituting (4.13) in (4.12) with (4.14), (4.15) and neglecting friction forces $f_r\approx f_u\approx 0$, we get the following equation of the radial equilibrium:

$$w\frac{\partial w}{\partial r}-w_s^2\left[x\cos\theta+\frac{\partial\ln w_s}{\partial s}\sin\theta-\frac{\operatorname{ctg}^2\beta}{r}+\right.$$
$$\left.+\operatorname{tg}\delta\left(\frac{\sin\theta\operatorname{ctg}\beta}{r}+\frac{\partial\operatorname{ctg}\beta}{\partial s}-\frac{\partial\ln w_s}{\partial s}\operatorname{ctg}\beta\right)\right]+$$
$$+2\varpi w_s\left(\operatorname{ctg}\beta-\sin\theta\operatorname{tg}\delta\right)=\frac{\partial H}{\partial r}-T\frac{\partial s}{\partial r}. \tag{4.16}$$

Turning to the new independent variable ψ – the stream function, we write (4.16) in final form:

$$w'=\left(\mu r\rho\cos\theta\right)^{-1}\left\{\left[x\cos\theta+\frac{\partial\ln w_s}{\partial s}\sin\theta-\frac{\operatorname{ctg}^2\beta}{r}+\right.\right.$$
$$\left.+\frac{\sin\theta\operatorname{ctg}\beta\operatorname{tg}\delta}{r}+\frac{\partial\operatorname{ctg}\beta}{\partial s}\operatorname{tg}\delta+\frac{\partial\ln w_s}{\partial s}\operatorname{ctg}\beta\operatorname{tg}\delta\right]\sin\beta-$$
$$\left.-2\frac{\varpi}{w}\left(\operatorname{ctg}\beta-\sin\theta\operatorname{tg}\delta\right)\right\}+\frac{1}{w}\left(H'-TS'\right). \tag{4.17}$$

Equation (4.17) for given geometrical parameters of the surface S_2' forms a closed system of ordinary differential equations in cross-section $z=\text{const}$ together with the continuity equation:

$$r'=\left(\mu r\rho w_s\cos\theta\right)^{-1}. \tag{4.18}$$

Consider a three sections stage calculation which located on the entrance and exit edges of the guide vane and on the trailing edge of the impeller. Derivative $\partial\ln w_s/\partial s$ is defined in terms of the flow of the working fluid in the free space (right side of the design section):

$$\frac{\partial\ln w_s}{\partial s}=\frac{1}{\mathrm{M}_s^2-1}\left[\frac{\cos\theta}{r}\frac{\partial\left(r\operatorname{tg}\theta\right)}{\partial r}-x\operatorname{tg}\theta+\frac{\mathrm{M}_u^2}{r}\sin\theta\right]. \tag{4.19}$$

In the absence of lean $\left(\operatorname{tg}\delta=0\right)$ the equation (4.17) coincides with the previously obtained. Upheld algorithm for the stage calculation by sections and supplements it by specifying the lean angles of the guide and rotor blades output edges. Agreed $\delta(r) = \text{const}$.

Of particular interest is the study of the guide vane lean in order to assess its effect on the degree of reaction gradient and as a consequence, the amount of leakage in the radial space over rotor blades for different types of guide vane twists $\beta(r)$.

To determine $\partial \operatorname{ctg} \beta / \partial s$ the profile's backbone line appears as a parabola $u = u(z) = az + bz^2$, that gives approximately:

$$\frac{\partial \operatorname{ctg} \beta}{\partial s} \approx \frac{\operatorname{ctg} \beta_2 - \operatorname{ctg} \beta_1}{H}, \qquad (4.20)$$

where H – the width of the blade.

Estimated study shows that the lean most strongly affects the flow conditions in the stage just because a member $\dfrac{\partial \operatorname{ctg} \beta}{\partial s} \operatorname{tg} \delta$. If there δ is positive, an alignment of reaction gradient happens, and, as can be seen from (4.20), the stronger, the smaller the α_1 angle and narrower the blade. Analysis (4.17) shows that the pressure compensation in the gap due to nozzle reverse twist occurs because of the appearance of the curvature of the stream lines in the gap, i.e. this influence is indirect and requires specification of the form of the stream lines in the stage calculation.

On the contrary, the effect of the lean angle on the degree of reaction gradient in the equation (4.17) manifests itself through the curvature of the surface in the oblique cut area, which allows the stage calculation even within a cylindrical theory.

Calculations were performed to determine the effect of the nozzle tangential lean on the characteristics of the experimental air turbine stage with high load $D_m/l = 14$, with the value of the radial gap $\delta_r = 1.5$ mm.

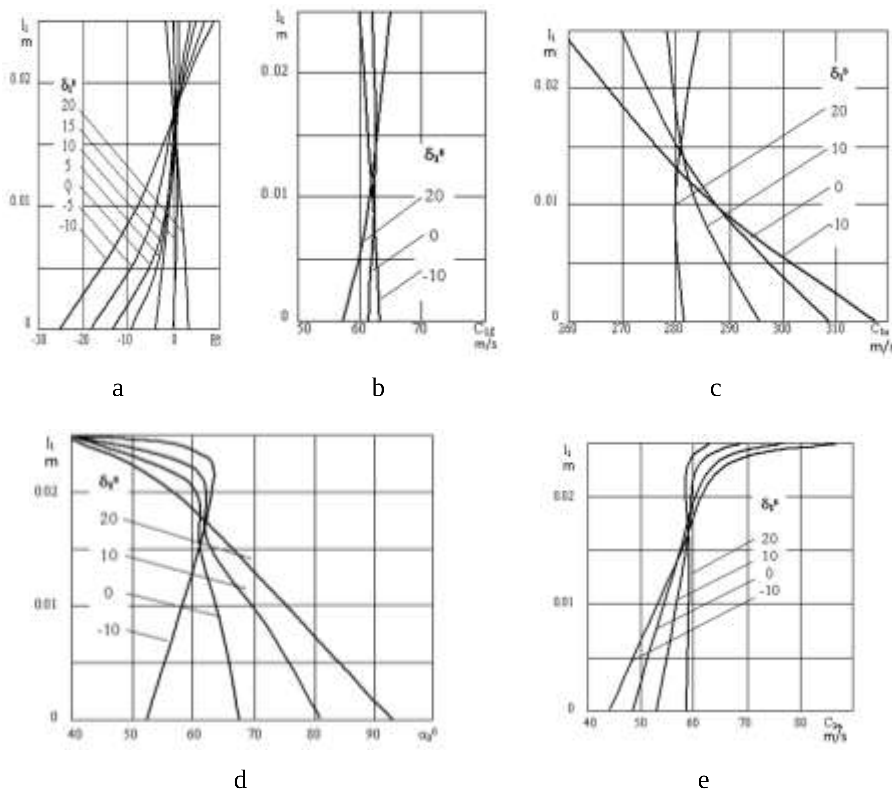

Figure 4.4 *The impact of the lean on the stage performance with m = 1.*

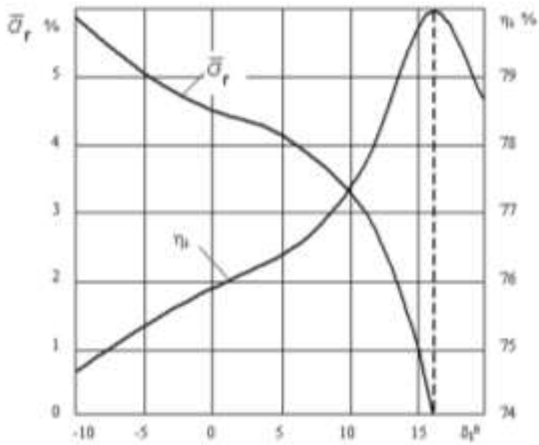

Figure 4.5 *Dependence of the radial clearance relative mass flow and stage internal efficiency on the lean angle for m = 1.*

Vane twist law was set in the form (4.4) and the lean angle ranged –10°…+20°. The results of calculations for m = 1 (the constant circulation law), and m=-8 (reverse twist, optimal when this value of radial clearance in the absence of the lean) are shown in Fig. 4.4, 4.5 and Fig. 4.6, 4.7. The figures show that the lean is a powerful tool in controlling the stage flow, in some cases giving an opportunity to significantly increase its efficiency.

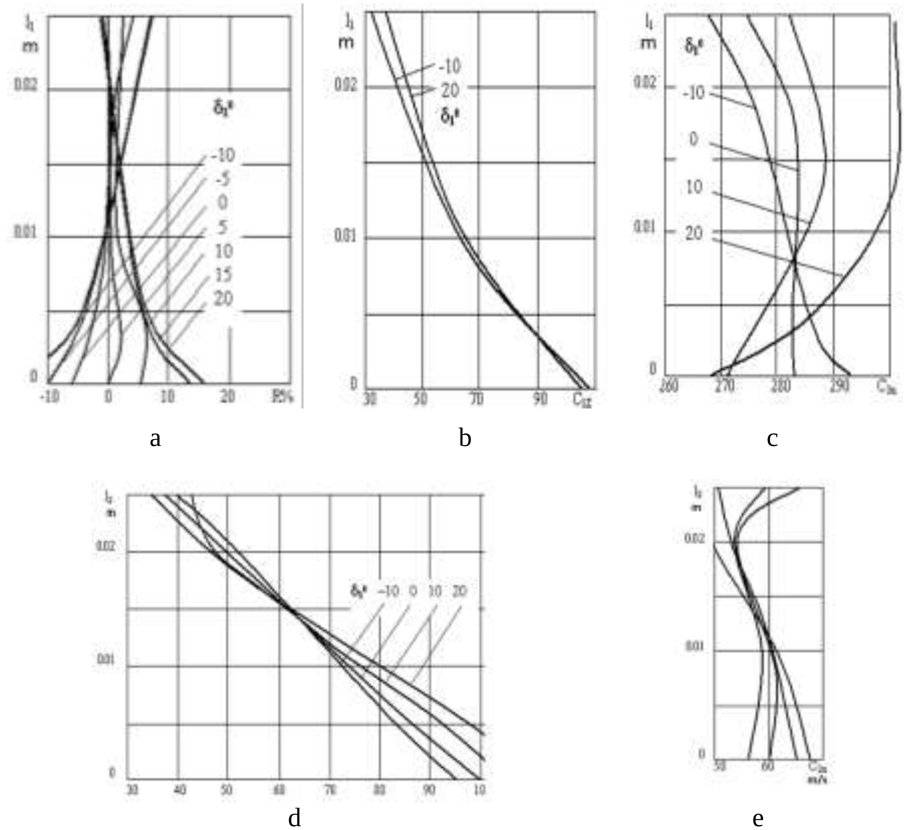

__Figure 4.6__ The impact of the lean on the stage performance at m = –8.

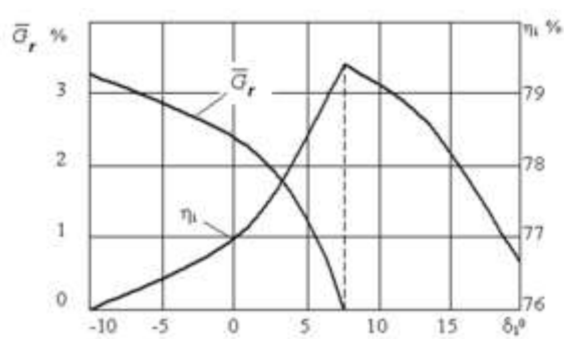

__Figure 4.7__ Dependence of the relative mass flow in the radial clearance and stage internal efficiency on the lean angle at m = –8.