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ANDRZEJ KORCZAK[a*], ADAM PAPIERSKI[b]

The flow through the face clearance of the disk relieving the axial force in multi-stage centrifugal pump

[a] *Institute of Power Machinery, Silesian University of Technology, 44-101 Gliwice, Konarskiego 18, Poland*

[b] *Institute of Turbomachinery, Technical University of Łódź, Wólczańska 219/223, Poland*

Abstract

The paper deals with the turbulent flow through the face clearance, which is a characteristic element of the flow system of the disk which relieves the axial force in a multi-stage centrifugal pumps. The presented description illustrates the automodelling flow, i.e. the square dependence of hydraulic resistances on the flow rate, applying the model k-ε and making use of the Tasc-Flow program of calculations. The essential aim was to compare the pressure fields in both these descriptions. The pressure field decides about the value and direction of the hydrodynamic moment affecting the walls of the clearance. The hydrodynamic forces and moments in the sealing clearances influence essentially the dynamic properties of the rotor of centrifugal pumps. The results of calculations have been compared basing on a numerical example with the same assumed geometry of the clearances, the same flow rate and the same pressure drop. The pressure fields obtained by means of both methods of calculations have been presented graphically, providing numerical values of the calculated axial forces and moments, as well as the direction of their activity. Moreover, for the selected cross-section of the clearance the fields of radial and spherical speeds have been given. The coefficients of the inlet resistance and continuous resistance in the clearance, the application of which in the description of the automodelling flow provided the pressure field have been calculated similarly as in the case of applying the model k-ε . As we see, in order to obtain correct results, the description of the automodelling flow ought to differentiate the coefficient of continuous resistance along the perimeter of the clearance.

Keywords: Turbulent flow; Face clearance

*Corresponding author. E-mail address: korczak@rie5.ise.polsl.gliwice.pl

Nomenclature

a, a_m	–	width of the clearance, mean value of this width, m
F, F_a	–	force, axial force, N
M, M_x, M_y	–	momentum, components of the momentum vector, Mm
p, p_2, p_3	–	pressure, pressure at the inlet, pressure at the outlet of the clearance, Pa
$\Delta p, \Delta p_{wl}, \Delta_{sz}$	–	total, inlet and continuos pressure drop in the face clearance, Pa
q	–	flow rate through the clearance, m^3/sec
r, φ, z	–	coordinates of the cylindrical coordinate system, m, rad, m
r_2, r_3	–	internal radius, external radius of the clearance, m
Re, Re_P, Re_C	–	Reynolds similarity number, Re in Poiseuille's flow, Re in Couette's flow
$u = \omega r$	–	circumferential speed of the wall, m/s
U_j $(j, i = 1, 2, 3)$	–	components of the vector of the flow velocity in a Cartesian coordinate system (in T-F), m/sec
v_r, v_φ, v_z	–	components of the vector of the flow velocity in a cylindrical coordinate system, m/sec
x, y, z	–	coordinates of a Cartesian coordinate system, m
$x_{j(i,j=1,2,3)}$	–	coordinates of a Cartesian coordinate system, m
α	–	angle of convergence of the walls of the clearance, rad
ε	–	value of energy of dissipation (in T-F), m^2/s^2
λ	–	coefficient of continuity resistance
μ	–	dynamic viscosity, kg/ms
μ_T	–	dynamic turbulent viscosity (in T-F), kg/ms
$\nu = \mu\rho$	–	kinematic viscosity, m^2/sec
ω	–	angular velocity of the rotating ring, s^{-1}
$\vec{\omega}$	–	angular velocity of the rotating system (in T-F), s^{-1}
ρ	–	density of liquid, kg/m^3
ψ	–	coefficient part of inlet losses
ϑ	–	load capacity coefficient of slip ring
ζ	–	coefficient of inlet resistance

1 Introduction

The face clearance is an essential element of the flow system of hydraulic machines, particularly of balance disks (Fig. 1), the sealings of the rotor necks of the rotors of single stage pumps for contaminated liquids, but also of face seals. Whereas in the relieving disks and sealings of the rotor necks we have mostly to do with turbulent or transient flows, in sliding face seals there occur either boundary or laminar flows. In rigidly fastened sliding rings and stop collars of the balance disks the distribution of pressure in the face clearance does not essentially affect the operation of this structural node.

It works correctly if dry friction between the rings is prevented, warranting an adequate width of the clearance [5]. The application of a flexible slip-ring or stop collar permits to take advantage of the favourable distribution of pressure in the

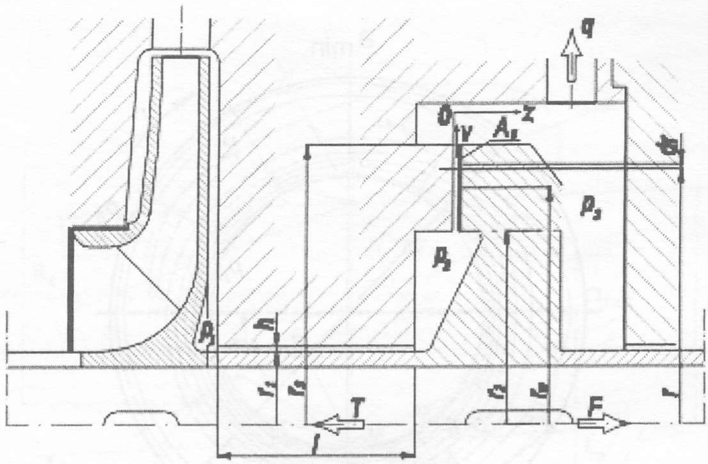

Figure 1. Balance disk of a multi-stage centrifugal pump with a characteristic face clearance.

face clearance, which is the case when the flow is centrifugal, so that the walls of the clearance do not contact. In this way the fluid friction can be maintained in spite of the average width of the clearance being several times smaller than the sum of the axial run-out of both walls of the clearance [1].

Figure 2 provides exemplary results of measurements of the pressure in the face clearance [3]. These measurements were taken for: $r_2 = 80$ mm, $r_3 = 140$ mm, $a_m = 0.25$ mm, $\Delta a = 0.1$ mm, $p_2 - p_3 = 0.46$ bar, $q = 0.87$ l/s, $n = 2820$ r.p.m.

The distribution of pressure in the face clearing with an assumed axial run-out can be measured in a limited range of its dimensions and parameters. It is particularly difficult to achieve a controlled width of the clearing within the range < 0.1 mm, while the slip-ring rotates. Large pressure drops are also restricted by the rigidity of the model, imposing the range of the parameters, in particular the range of average clearance widths and pressure drops, as assumed, for instance, in [3]. In the balance disks of multi-stage centrifugal pumps, employed e.g. to feed steam boilers, the pressure drop in the face clearance may amount to 10 MP, the width of the clearance amounting to < 0.1 mm.

In general, the actual level of technology permits to obtain butting faces of the slip-ring and stop collar featured by negligible out-of-flatness and deviation of its smoothness. Operational tests have proved that losses caused by erosion, amounting even to several millimeters of the wall thickness do not affect the smoothness of the walls, if the hardness of the material of the rings is adequately differentiated. The eccentricity of the rings constituting the face clearance may be neglected, because it must be related to the length of the clearance.

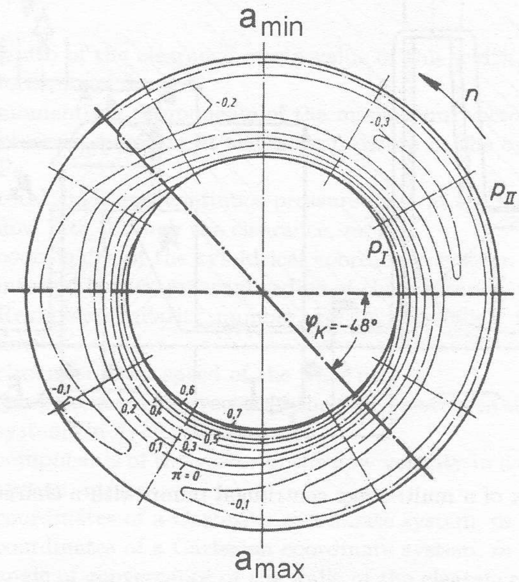

Figure 2. Measured pressure distributions in the face clearing [3].

Of essential importance for the flow, and particularly also for the distribution of pressure in the face clearance, is the convergence resulting from the axial run-out of both surfaces, due to the dimensional tolerances and coaxial arrangement of the shaft and the casing of the pump. This is also affected by the static deflection and precession of the rotor. The parallelity error of the walls in the face clearance of the relief unit is practically unavoidable. The geometry of the face clearance has been presented in Fig. 3.

In the present paper it has been assumed that the stop collar is perpendicular to the axis of the shaft, i.e. the angles $\alpha_0 = \varphi_0 = \varphi_m = 0$, and the slip-ring is tilted at an angle of α_s. The erosion of the slip-ring and the stop collar occurring in the balance disk due to the operation of the pump does not influence the geometry of the clearance. The increase of its average width and of the flow rate through the relief unit (Fig. 1) is caused by the erosion of the longitudinal clearance in the outlet casing of the pump.

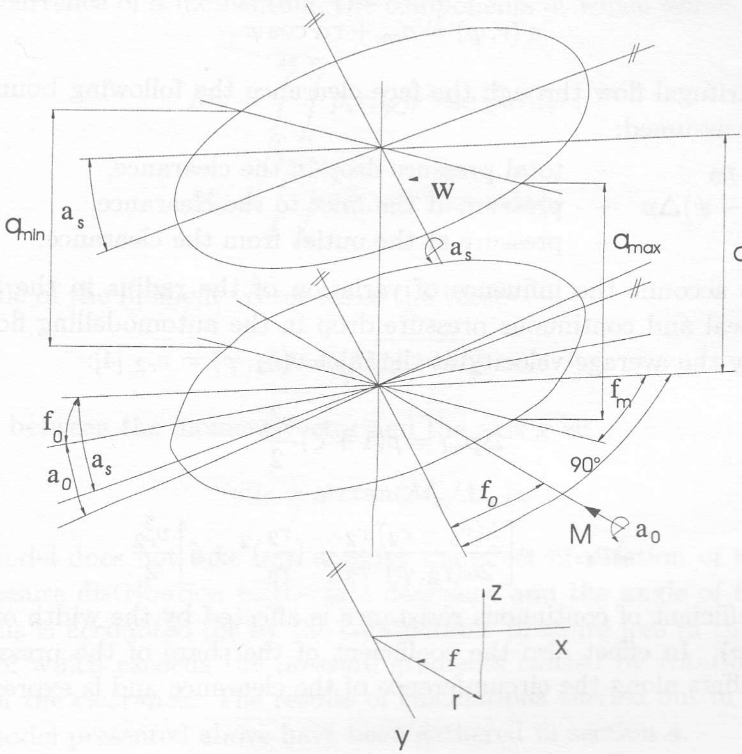

Figure 3. Geometry of the face clearance.

2 Description of the automodelling flow through the face clearance (Fig. 3)

In fundamental calculations of the flow through throttling clearances it has been assumed that the drag coefficient is independent of the Reynolds number and that we have to do with an automodelling flow. In the case of large gradients, when the kinetic energy in the face clearance is neglected the equation of motion in the radial direction takes the following form [2, 5]:

$$\frac{d\,(pa)}{2dr} = -\tau_r \, .$$

Taking into account the Darcy-Weisbach's empirical formula as well as the averaged condition of continuous flow we get:

$$d\,(pa) = -B\frac{dr}{r^2} \, . \tag{1}$$

The width of the face clearance at any arbitrary point is expressed by the formula:

$$a\,(r,\varphi) = a_m + r\alpha\cos\varphi\ .$$

For the centrifugal flow through the face clearance the following boundary conditions were assumed:

$\Delta p = p_2 - p_3$ – total pressure drop in the clearance,
$p(r_2) = (1-\psi)\Delta p$ – pressure at the inlet to the clearance,
$p(r_3) = p_3$ – pressure at the outlet from the clearance.

Taking into account the influence of variation of the radius in the face clearance, the local and continuous pressure drop in the automodelling flow may be expressed by the average velocity at the inlet $v(r_2,\varphi) = v_{r2}$ [4]:

$$\Delta p_{wl} = \rho(1+\zeta)\frac{v_{r2}^2}{2} \tag{2a}$$

and

$$\Delta p_{sz} = \rho\Big[\frac{\lambda(r_3-r_2)}{2a(r_2,\varphi)}\frac{r_2}{r_3} + (\frac{r_2}{r_3})^2 + \zeta\Big]\frac{v_{r2}^2}{2}\ . \tag{2b}$$

The coefficient of continuous resistance is affected by the width of the clearance $a(r_2,\varphi)$. In effect also the coefficient of the share of the pressure loss at the inlet differs along the circumference of the clearance and is expressed by the formula:

$$\psi\,(\varphi) = \frac{1+\zeta}{\frac{\lambda(r_3-r_2)}{2a(r_2,\varphi)}\frac{r_2}{r_3} + (\frac{r_2}{r_3})^2 + \zeta} = \frac{\Delta p_{wl}}{\Delta p_{sz}+\Delta p_{wl}} = \frac{\Delta p_{wl}}{\Delta p}\ . \tag{3}$$

In our example in section 4, it was $\zeta = 0.5$ and $\lambda = 0.04$.

After integration of equation (1) and calculations of the coefficient B and integration constant, basing on the geometry of the clearance and boundary conditions, the obtained pressure value in any arbitrary point of the clearance amounted to:

$$p\,(r,\varphi) = (1-\psi)\,\Delta p\frac{a_2}{a}\frac{r_2}{r}\frac{r_3-r}{r_3-r_2}\ . \tag{4}$$

The resultant axial force is expressed by the formula:

$$F_z = \int\limits_{r_2}^{r_3}\int\limits_{0}^{2\pi} p(r,\varphi)rd\varphi dr\ . \tag{5}$$

The load capacity coefficient of the slip-ring is expressed by the quotient:

$$\vartheta = \frac{F_z}{\pi(r_3^2-r_2^2)\Delta p}\ . \tag{6}$$

The lack of a symmetrical distribution of pressure in the clearance is responsible for the occurrence of a momentum, the components of which are:

$$M_x = \int_0^{2\pi}\int_{r_1}^{r_2} p(r,\varphi) r^2 \sin\varphi dr d\varphi \ ,$$
$$M_y = \int_0^{2\pi}\int_{r_1}^{r_2} p(r,\varphi) r^2 \cos\varphi dr d\varphi \ . \tag{7}$$

The module of the moment vector takes the value:

$$M = \sqrt{M_x^2 + M_y^2} \ . \tag{8}$$

The angle between the moment vector and the axis x is:

$$\varphi_M = \arctan(M_y/M_x) \ . \tag{9}$$

This model does not take into account the effect of rotation of the slip-ring on the pressure distribution in the face clearance and the angle of the moment vector. This is accounted for by the considerable pressure loss in the clearance, the order of which exceeds the pressure gradients caused by rotation of one of the walls of the clearance. The results of calculations carried out in compliance with the model presented above have been gathered in section 4.

3 Calculation model of k-ε applying the Tasc Flow programme

In order to determine the structure of a three-dimensional flow the commercial programme CFX-Tasc Flow developed by ASA Technology Engineering Software Company [10] was applied.

3.1 Basic equations

The basic equations in the assumed system of Cartesian co-ordinates look as follows:

- equation of continuity:

$$\frac{\partial}{x_j}(\rho U_j) = 0 \, , \tag{10}$$

where U_j stands for the Cartesian components of the three-dimensional velocity vector $(j, i) = 1, 2, 3$;

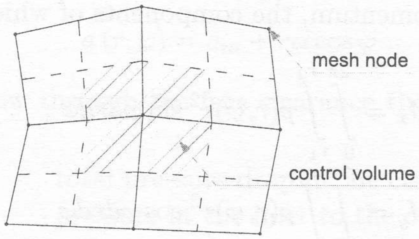

Figure 4. Arrangement of the elementary control volume in the calculation net for the integration of basic equations.

- equation of the conservation of motion:

$$\frac{\partial}{\partial t}(\rho \overline{U}_i) + \frac{\partial}{\partial x_j}(\rho \overline{U}_i \overline{U}_j) = -\frac{\partial \overline{p}}{\partial x_i} - \frac{\partial}{\partial x_j}\left\{\mu_{eff}\left(\frac{\partial \overline{U}_i}{\partial x_j} + \frac{\partial \overline{U}_j}{\partial x_i}\right)\right\} + f_i, \quad (11)$$

where the term $f_i = -\rho\left(2\vec{\omega} \times \vec{U} + \vec{\omega} \times (\vec{\omega} \times \vec{r})\right)$ represents in the system rotating with the angular velocity $\vec{\omega}$, the Coriolis force and the centrifugal force, whereas $\mu_{eff} = \mu + \mu_T$, where μ_T denotes the turbulent viscosity determined by means of the selected model of turbulence.

The authors have noticed that calculations of the flow in this type of clearances with a discretization of the first order yielded results much different from the results obtained experimentally and by means of other methods of calculations. This is caused by the occurrence of the so-called numerical viscosity in schematic diagrams of the first order [6].

3.2 Model of turbulence

The value of the term of Reynolds stresses is rather unknown, and if so, only in simple cases. The value of these stresses is connected with the averaged distribution of the velocity fields. The relations between the velocity field and the stresses are final equations and have been called models of turbulence. In the investigations presented in this paper the turbulent viscosity μ_T was determined making use of the $k - \varepsilon$ turbulence model.

Turbulent viscosity is proportional to the product of the turbulent scale of velocity v_t and the way of mixing l_t, defined by Prandtl, viz:

$$\mu_T = \rho \cdot c_\mu \cdot v_t \cdot l_t \,. \quad (12)$$

In the model k-ε the value v_t, is determined as the root of the kinetic energy of the turbulence $v_t = \sqrt{k}$. The way of mixing l_t is connected with the equation $\varepsilon = \frac{k^{3/2}}{l_t}$. Substituting it, we get the relation concerning the dynamic turbulent viscosity $\mu_T = \rho \cdot c_\mu \frac{k^2}{\varepsilon}$, whereas k and ε are determined by means of the following differential equations:
for k:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho U_j k)}{\partial x} = P_k - \rho\varepsilon + \frac{\partial}{\partial x_j}\left(\frac{\mu_T}{\sigma_k}\frac{\partial k}{\partial x_j}\right), \tag{13}$$

for ε:

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \frac{\partial(\rho U_j \varepsilon)}{\partial x} = \frac{\varepsilon}{k}(C_{\varepsilon 1} P_k - C_{\varepsilon 2}\rho\varepsilon) + \frac{\partial}{\partial x_j}\left(\frac{\mu_T}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x_j}\right) \tag{14}$$

where P_k denotes the production of the energy of turbulence:

$$P_k = \mu_T\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j}. \tag{15}$$

The constants in this model are:

$$C_\mu = 0.09 \qquad C_{\varepsilon 1} = 1.44, \qquad C_{\varepsilon 2} = 1.92, \qquad \sigma_k = 1.0, \qquad \sigma_\varepsilon = 1.3\,.$$

They have been assumed basing on a preliminary analysis of the results of calculations of the flow through the face clearance, applying simplified formulae [2, 4] and comparing them with the results of measurements [1].

In the case of fluid-flow machines the frequently employed model described above requires the application of the so-called wall law in which a logarithmic distribution of velocities between the first node of the net of calculations is assumed, situated in the flow range and the node on the channel wall.

In the considered model of the flow through the face clearance the value of the Reynolds number may be in the range of a laminar, transient and turbulent flow. Within this range the logarithmic velocity distribution is of no importance. Therefore, the double-layer model $k-\varepsilon$ has been applied, in which the flow range has been divided into nodes in which $\mu_T/\mu \geq 36.$ and nodes in which $\mu_T/\mu < 36.$ This delimiting criterion had been suggested by Rodi [9].

In the external layer the turbulent viscosity is determined by means of the double-equation standard model $k - \varepsilon$. And in those nodes in which $\mu_T/\mu < 36$, the viscosity is determined by means of a single-equation model, in which Eq. (13) must be solved. The value of the energy of dissipation ε is determined by means of the equation:

$$\varepsilon = \frac{k^{3/2}}{\ell_t \cdot f_\varepsilon}$$

and the turbulent viscosity by:

$$\mu_T = \rho \cdot C_\mu \sqrt{k} \cdot \ell_t \cdot f_\mu$$

where the way of mixing:

$$\begin{array}{lcl} \ell_t & = & \frac{\kappa \cdot y}{C_\mu^{3/4}} \\ \kappa = 0.4 & - & \text{von Karman's constant} \\ \text{y} & - & \text{distance between the node and the wall} \end{array}$$

and the function of influence:

$$f_\varepsilon = 1 - \exp\left(-\frac{y^+}{3.8 \cdot C_\mu^{1/4}}\right) \quad f_\mu = 1 - \exp\left(-\frac{y^+}{63 \cdot C_\mu^{1/4}}\right)$$

in which y^+ is the dimensionless distance between the node and the wall.

Of essential meaning is the control of the value y^+ in the course of calculations. If the calculations are carried out by means of the model k-ε, the value of y^+ ought to be contained within the range of 20-100, and in the case of the double-layer model $y^+ < 2$. If these conditions are not satisfied, the calculation mesh must be modified.

3.3 Boundary conditions of calculations by means of the TascFlow code

In the calculations the following boundary conditions were given:

- value of the total pressure in the inlet chamber of the clearance,
- value of the static pressure in the outlet chamber of the clearance,
- neutralization of velocities on the immobile walls of the clearance and both chambers,
- the values of peripheral velocities on the mobile walls of the clearance and the chambers,
- the intensity of turbulence and the way of mixing concerning the $k - \varepsilon$ turbulence model in the inlet chamber of the clearance.

The applied numerical values were identical with those applied in the description of the automodelling flow.

4 Example of calculations

This paper provides a comparison of the results of calculations concerning the distribution of pressure $p(r, \varphi)$ in the face clearance, the axial force F_z, coefficient of this force ϑ, momentum M and its components M_x, M_y, the angle of the moment vector φ_M, carried out basing on the assumption of automodelling flows and the description by means of the model $k - \varepsilon$. Moreover the field of radial velocity and the field of peripheral velocity along the radius for the selected peripheral angle calculated by means of the model $k - \varepsilon$ have been presented. These calculations were carried out assuming the same geometry of the face clearance and the same values of pressure at the inlet and the outlet.
Hereby the following assumptions were made: $r_2 = 127.5$ mm, $r_3 = 170$ mm, $a_m = 0,12$ mm, $\alpha_s = 0.00047$ rad, $\alpha_0 = 0$, $\omega = 157\,\mathrm{s}^{-1}$. Parameters of the flowing water: $p_2 = 5.5$ MPa, $p_3 = 0.1$ MPa, temperature $t = 15°$C, $\nu = 1.1414$ m^2/s, $q = 0.00423$ m^3/s, the resulting mean velocity amounting to $v_{rm} = 37.71$ m/sec.
Average criterial Reynolds numbers in:

- Poiseuille's flow $\mathrm{Re}_p = \frac{2sv_{rm}}{\nu} = 7930...$
- Couette's flow $\mathrm{Re}_c = \frac{2\omega r_m a_m}{\nu} = 4750.$

5 Results of calculations

Fundamental results of calculations carried out in compliance with both descriptions are the pressure fields concerning the varying peripheral angles and the radius of the clearance over its entire surface, as shown in Figs. 5 and 6. Due to simplifications applied in the description similarly as in the case of automodelling flows the curves of pressures are symmetrical to the axis of symmetry of the clearance, i.e. the axis Ox.

Calculations carried out according to the programme Tasc Flow provided also curves of the radial and circumferential velocities in the clearance, as shown in Figs. 7 and 8 concerning the selected longitudinal section. The maximum value of the radial velocity in the clearance amounted to $v_{rmax} = 59.4$ m/sec. The velocity profiles are monotonic after they have been formed by turbulences caused by the resistance at the inlet. The case looks similar concerning other longitudinal sections of the clearance.

Besides simple cases, the value of Reynolds stresses is unknown. The values of such stresses are related to the averaged distribution of velocity fields. The relation existing between the velocity field and the stresses are final equations, called models of turbulence. In the case of simple models empirical values of these relations have been obtained for a given geometry of the flow.

A comparison of the numerical values obtained according to both models of

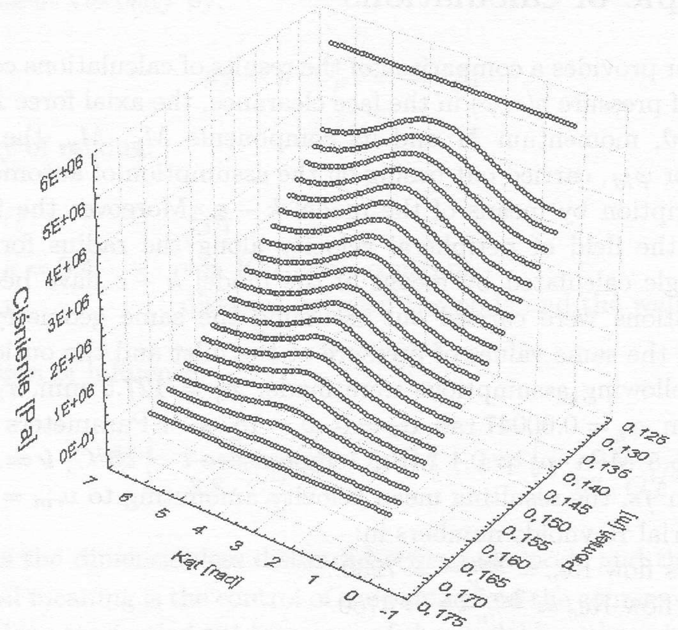

Figure 5. Distribution of pressures calculated similarly as in the case of automodelling flow.

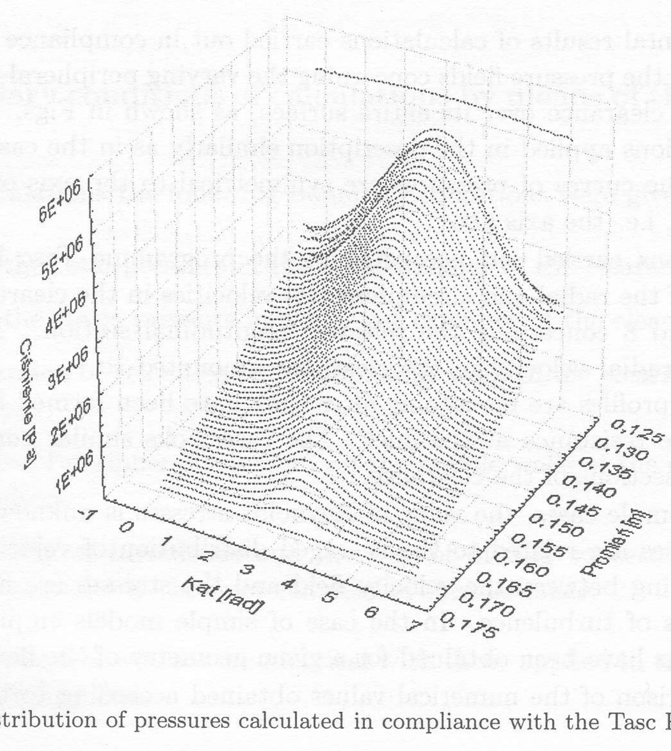

Figure 6. Distribution of pressures calculated in compliance with the Tasc Flow code.

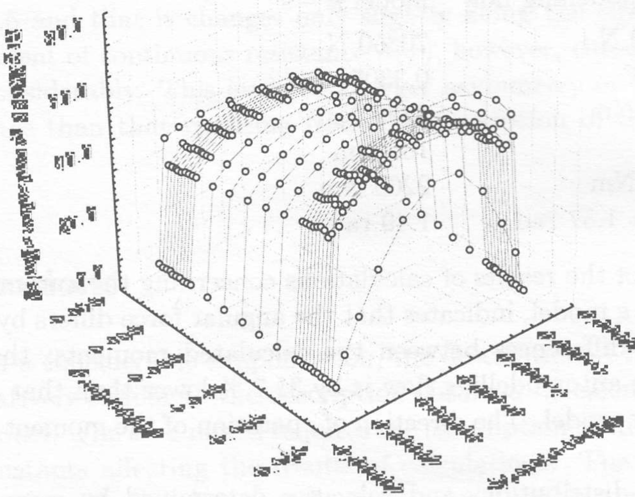

Figure 7. Distribution of the radial velocity in the clearance calculated in compliance with the Tasc Flow code, concerning a constant circumferential angle φ=183^0.

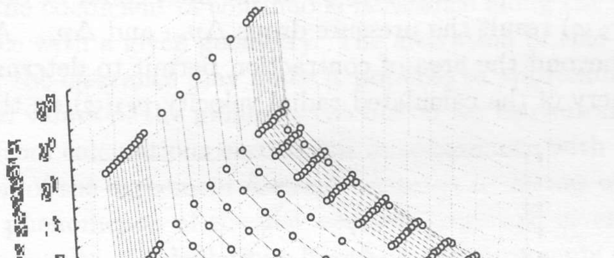

Figure 8. Distribution of the peripheral velocity in the clearance, calculated in compliance with the Tasc Flow code, concerning a constant circumferential angle φ=183^0.

the flow through the face clearance:

		automodelling flow	model $k-\varepsilon$
F_z	=	77730 N	70850 N
ϑ	=	0.3626	0.3305
M	=	1543 Nm	2270 Nm
M_x	=	0	166 Nm
M_y	=	1543 Nm	2260 Nm
φ_M	=	$\pi/2 = 1.57$ rad	1.49 rad

The comparison of the results of calculations concerning the automodelling flow, basing on the $k-\varepsilon$ model, indicates that the angular force differs by 8.8 %. More essential are the differences between the calculated moments; the momentum calculated for the automodelling flow is by 31.7 % lower than that calculated by means of the $k-\varepsilon$ model. The direction of operation of the moment vector differs only slightly.

The pressure distributions and velocities determined by means of the $k-\varepsilon$ model make it possible to calculate the coefficients of inlet losses $\zeta(\varphi)$ and continuity resistance $\lambda(\varphi)$; when these are substituted in the equations (2a) and (2b), we get the curves of pressure in the clearance, as calculated by means of the model $k-\varepsilon$. Linearizing the curve of pressure in the clearance obtained basing on the $k-\varepsilon$ model, and extending it across the area of contraction as far as the inlet, the calculated pressure $p(r_{2,}\varphi)$ at the inlet could be determined. From this pressure $p(r_{2,}\varphi)$ result the pressure drops Δp_{wl} and Δp_{sz}. Also the velocity profiles formed beyond the area of contraction permit to determine their course along the periphery of the calculated radial velocity $v_{r2}(\varphi)$ on the radius r_2.

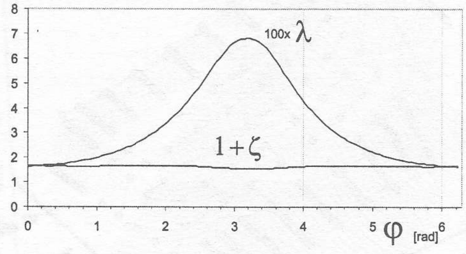

Figure 9. Curves of the coefficients of the inlet resistance $1+\zeta(\varphi)$ and the continuous resistance $\lambda(\varphi)$ in the clearance, calculated by means of the equations (2a) and (2b), after substituting in them the quantities Δp_{wl} and Δp_{sz} taken from the description of the $k-\varepsilon$ model.

For the pressure drops Δp_{wl} and Δp_{sz} and inlet velocities $v_{r2}(\varphi)$ determined by means of the equations (2a) and (2b), the coefficients of inlet resistance ζ and continuity resistance λ have been calculated and presented in Fig. 9.

The diagram presented in Fig. 9 shows that the coefficient of inlet losses calculated in this way does not deviate much from the generally assumed value $1 + \zeta(\varphi) = 1.5$ and that it changes only slightly along the circumference. The average coefficient of continuous resistance $\lambda(\varphi)$, however, changes along the circumference considerably. This leads to a larger asymmetry of the pressure field in the clearance than that resulting from the description of the automodelling flow.

6 Conclusions

In spite of a considerable simplification, the description of the automodelling flow is qualitatively similar to the description obtained by calculations according to the $k-\varepsilon$ model. The $k-\varepsilon$ model requires the assumption of the values of several empiricial constants affecting the results of calculations. The determination of the flow rate through the clearance by means of the first algorithm, applying coefficients taken from literature and determined by measurements permits to verify the assumptions for calculations according to the $k - \varepsilon$ model.

Calculations carried out in compliance with the $k - \varepsilon$ model have proved that the axial force is not much smaller than that calculated for the automodelling flow, whereas the momentum is much greater. The results of TF calculations indicate a variability of the coefficient of continuous resistance along the circumference of the face clearance with a given geometry. The alteration of this coefficient along the periphery of the clearance may make it possible to approximate the pressure distributions and values of the moments calculated for the automodelling flow to those resulting from calculations carried out in compliance with the $k - \varepsilon$ model. The confirmation of the agreement of the description by means of the $k - \varepsilon$ model with the actual phenomenon of the flow requires empirical investigations. Thus, for instance, the results of calculations based on measurements of the coefficient of the inlet resistance, quoted in [3], indicate that values between 0.1 and 0.2 have been assumed, i.e. smaller ones than those obtained in our analysis.

The application of coefficients obtained in the quoted numerical example in the case of a clearance with other dimensions and other parameters of operation may make it possible to assess quantitatively the results obtained similarly as in the case of the automodelling flow.

As most of the results of investigations concerning the flow through the clearance, described in literature, are based on the assumption of the automodelling flow, this description may contribute to the assessment of the flow rate and turbulence of the flow, which – in turn – will permit to specify more precisely the assumptions dealing with calculations in compliance with the $k - \varepsilon$ model.

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References

[1] Korczak A.: *Test stand for model balance disks relieving the axial forces in multistage centrifugal pumps*, 12th Sem. on Energy, Silesian University of Technology, Gliwice 2003 (in Polish).

[2] Marcinkovski V.A.: *Hydrodynamics of throttling flow channels*, Publ. Sumy State Univ., Sumy 2002 (in Russian).

[3] Kosyna G.: *Investigations concerning radial-flow sealing clearence with a flat plates, taking into account parallel deviations*, Doctor's Dissertation, Tech. Univ. Braunschweig 1976 (in German).

[4] Jędral W.: *Method of calculating the axial thrust and balancing devices in centrifugal pumps* (qualifying for proffesorship trial), Warsaw Univ. of Technology Publs., Mechanika, No. 110, Warsaw 1988 (in Polish).

[5] Tchegurko L. E.: *Balancing devices centrifugal boiler pumps in power station*, Energiya, Moscow 1978 (in Russian).

[6] Kazimierski Z.: *Numerical determination of turbulent flows*, Ossolineum, Wrocław 1992 (in Polish).

[7] Papierski A., Rabiega M.: *Multiblock paraller computation of an incompressible 3-D flow in turbomachines*, Task Quarterly 3, No. 1, (1999), 39-52.

[8] Chodkiewicz R., Papierski A.: *CFD code – a useful tool for the turbomachinery designer*, Sem./Summer School CFD for Turbomachinery Apl., 2001.

[9] Rodi W.: *Experience with two-layer models combining the model with a one-equation model near the wall*, Amer. Inst. of Aeronautics and Astronautics Paper 91-0216, Amer. Inst. of Aeronautics and Astronautics 29th Aerospace Sci. Meeting, Reno, USA, Jan. 7-10.1991.

[10] *CFX-TASCflow Theory Documentation. Version 2.12.*, AEA Technology Engineering Software Ltd. Waterloo, Canada.