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NEW VERSION OF THE UNIVERSAL MODELING FOR CENTRIFUGAL COMPRESSOR GAS DYNAMIC DESIGN

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ABSTRACT
The 4th generation model of the known TU SPb Universal modeling programs for gas dynamic design were perfect enough to predict design point efficiency with accuracy about 2,5% if a single set of coefficients was applied. To raise accuracy of calculations to 1% or less different sets of empirical coefficients were necessary for stages with different flow rate and work coefficients. The proposed text is focused on scientific background and realization of model improvements that leads to accuracy about 0,5% of design efficiency prediction with a single set of empirical coefficients for all types of stages and compressors. Samples of application are given.

Key words: compressor design, loss model, performance prediction, test data, model stages, flow rate coefficient, work coefficient.

Decades ago at pre – computer era design process consisted of empirically based set of rules application to choose main flow path dimensions. Serious model tests were obligatory before compressor manufacturing to check delivery pressure and efficiency. Better flow physical models and computer progress made possible to develop quickly operating programs to predict gas dynamic performance curves of an arbitrary flow path. TU SPb set of computer programs was named "The Universal modeling method" and its application still in mid 1990th had lead to elimination of model tests in a design process of industrial centrifugal compressors [1, 2, 3, 4, 5, 6]. Set of algebraic equations describe surface friction losses, flow separation and following mixing losses.

Flow deceleration along surfaces and velocity gradient along a normal to surfaces are taken into account. Schematically represented blade velocity diagram parameters are an important part of the model. Several dozens of compressor with delivery pressure up to 12,5 MPa, number of stages 1 - 8, power up to 25 mWt were designed for some Russian and foreign manufacturers by the preceding generation programs. Amount of compressor installed exceeds 400 with total power close to 5 000 000 kWt. In all cases the design parameters were achieved without model tests. The sample of typical 2-stage pipeline compressor is shown in the Fig. 1.

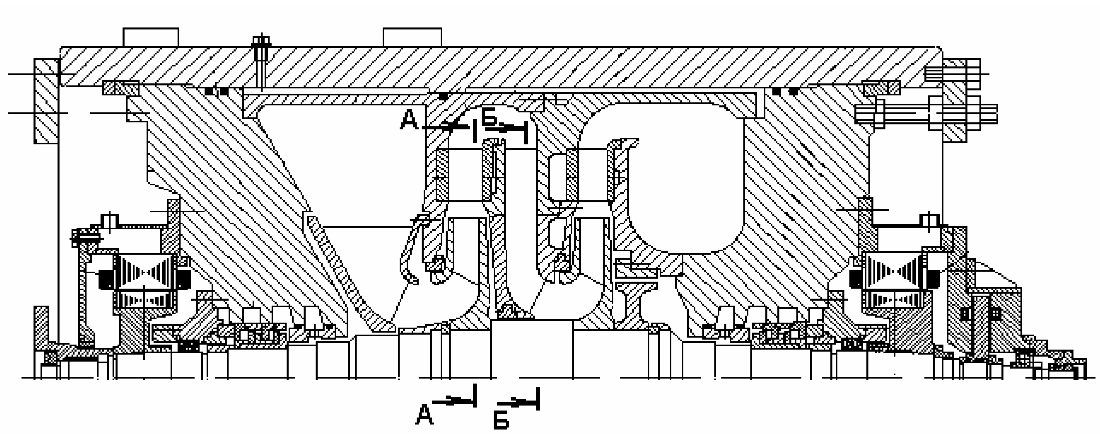

Fig. 1. Cross section of 2-stage pipeline centrifugal compressor

The previous models are able to predict design point efficiency with accuracy about 2,5% if a single set of coefficients was applied. There are necessary to apply different sets of coefficients for different types of impellers to increase prediction accuracy to 1%. The scientific background and realization of the new model with better abilities are presented in [7, 8, 9, 10]. New model for mixing mosses calculation was developed.

Surface roughness and shroud labyrinth seal leakage influence on flow at an impeller inlet were taken into account. One of important improvements was new blade velocity diagram schematization very close to non viscid diagram. (Two) Old and new ways of schematization are demonstrated by Fig. 2.

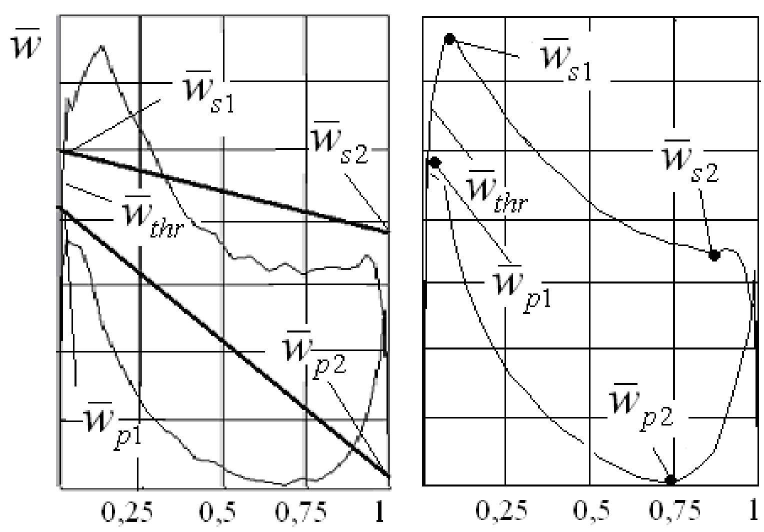

Fig.2. Non-viscid velocity diagrams along normalized length at a shroud blade-to-blade 2D impeller surface. Left – old simplified schematization (solid lines), right – new schematization by numerical experiment math reduction

The following equations demonstrate why characteristic velocities of a diagram w_{s1}, w_{s2}, w_{p1} w_{p2}, w_{thr1} shown in the Fig. 2 are important for performance curve(s) modeling. For instance, friction loss coefficient of impeller blades is equal:

$$\zeta_{fr\ prof} = \frac{2z}{\pi}\varepsilon_m \frac{\overline{S}_{bl}}{\Phi}\left(c_{ws}\left(\frac{w_s}{w_{thr1}}\right)^2 \overline{w}_s + c_{wp}\left(\frac{w_p}{w_{thr1}}\right)^2 \overline{w}_p \right). \tag{1}$$

There are members of Eq. 1 that include characteristic velocities:

$$c_{ws} = c_f\left(1 - X_i\left(4\frac{w_s/u_2}{R_{bl}/D_2}\right)^{X_i} + X_i\left(1 - \frac{w_{s2}}{w_{s1}}\right)^{X_i}\right), \tag{2}$$

$$c_{wp} = c_f\left(1 + X_i\left(\left(4\frac{w_p/u_2}{R_{bl}/D_2}\right)\right)^{X_i} + X_i\left(1 - \frac{w_{p2}}{w_{p1}}\right)^{X_i}\right), \tag{3}$$

and:

$$w_s = 0{,}5\left(w_{s1} + w_{s2}\right),\quad w_p = 0{,}5\left(w_{p1} + w_{p2}\right), \tag{4}$$

$$w_s^2 = 0{,}5\left(w_{s1}^2 + w_{s2}^2\right),\quad w_p^2 = 0{,}5\left(w_{p1}^2 + w_{p2}^2\right), \tag{5}$$

Drag force coefficient c_f in Eq. 1 is calculated by formulae for a thin plate with smooth or (of) rough surface [11];

$$c_f = X_i\frac{0.0307}{\mathrm{Re}_w^{1/7}} \text{ - hydraulically smooth surface,} \tag{6}$$

$$c_f = X_i\frac{1}{\left(1.89 + 1.62\cdot\lg\frac{1}{Rz/l_{bl}}\right)^{2.5}} \text{ - rough surface.} \tag{7}$$

The member $\left(4\frac{w_p/u_2}{R_{bl}/D_2}\right)$ in Eq. 2, 3 represents dimensionless velocity gradient normal to a surface. It influences a normal component of turbulent pulsations.

Characteristic velocities on a suction side are participating in calculation of mixing losses. The scheme of separation and mixing process are shown in the Fig. 3.

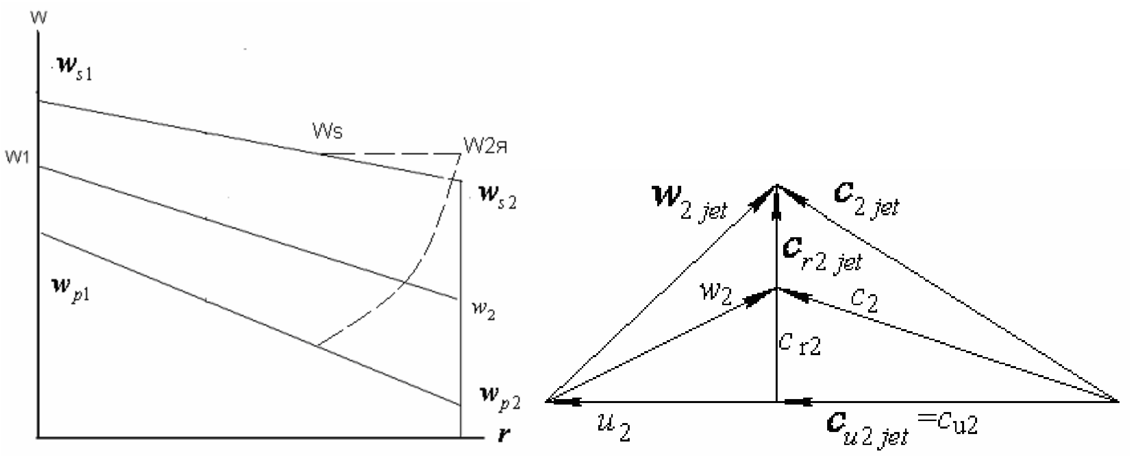

Fig. 3. Scheme of wake formation due to flow separation at a suction side (left) and flow mixing at an impeller exit (right)

The separation point position depends on normal velocity gradient that suppresses normal pulsations on a suction side of blades;

$$\dot{w}_s = \frac{w_s}{w_{s1}} = X_i\left(1 + X_i\left(4\frac{w_s/u_2}{R_{bl}/D_2}\right)^{X_i}\right). \tag{8}$$

Mixing loss coefficient corresponds to scheme of sudden expansion:

$$\zeta_{se} = X_i\left(\frac{w_{1з}}{w_1}\dot{w}_s \sin\beta_2 - \frac{c_{r2}}{w_1}\right)^2 . \tag{9}$$

Empirical coefficients values X_i ($i = 1-60$) depends on their position in presented above and other equations of the math model.

Models of 4th and earlier versions operate with simplified definition of characteristic velocities as shown in the Fig. 2 (left). Non-viscid velocity diagram is very close to a real one at design flow rates at least [12]. Simplified definition is based on a mean blade load. For instance in case of 2D impeller a mean load is equal:

$$\Delta\bar{w} = \frac{2\pi\tau}{z}\sin\frac{\beta_1+\beta_2}{2}\frac{\psi_T}{1-\bar{D}_1}. \tag{10}$$

Blade load distribution along a blade length (for any impeller) must be estimated by a program user arbitrarily and result is presented by solid lines in the Fig. 2 (left). To achieve principally better velocity diagram description the Authors applied results of numerical experiment presented at [7, 8]. About one hundred of impellers in range of flow rate coefficients 0,020 – 0,090, Euler work coefficients 0,45 – 0,80, relative hub 0,20 – 0,45 were designed and their velocity diagrams were calculated and mathematically reduced. The last version of the equation for maximum local velocity on a blade $\bar{w}_{s1}$ was proposed by one of the Authors:

$$\bar{w}_{s1} = \bar{w}_{thr1} + 0{,}5\left(2{,}06\frac{\bar{b}_1}{1-\bar{D}_1} - 0{,}261\psi_{T\,des} + 1{,}69\bar{D}_1^{2,8}\right)\Delta\bar{w}_{mean}\,. \tag{11}$$

Result of velocity diagram schematization by Eq. (11) and alike is demonstrated in Fig. 2 (right), (left).

Special attention was paid to more correct modeling of 3D impellers. Input menu of the 6th version program in the Fig. 4 shows more detailed description of an impeller meridian shape.

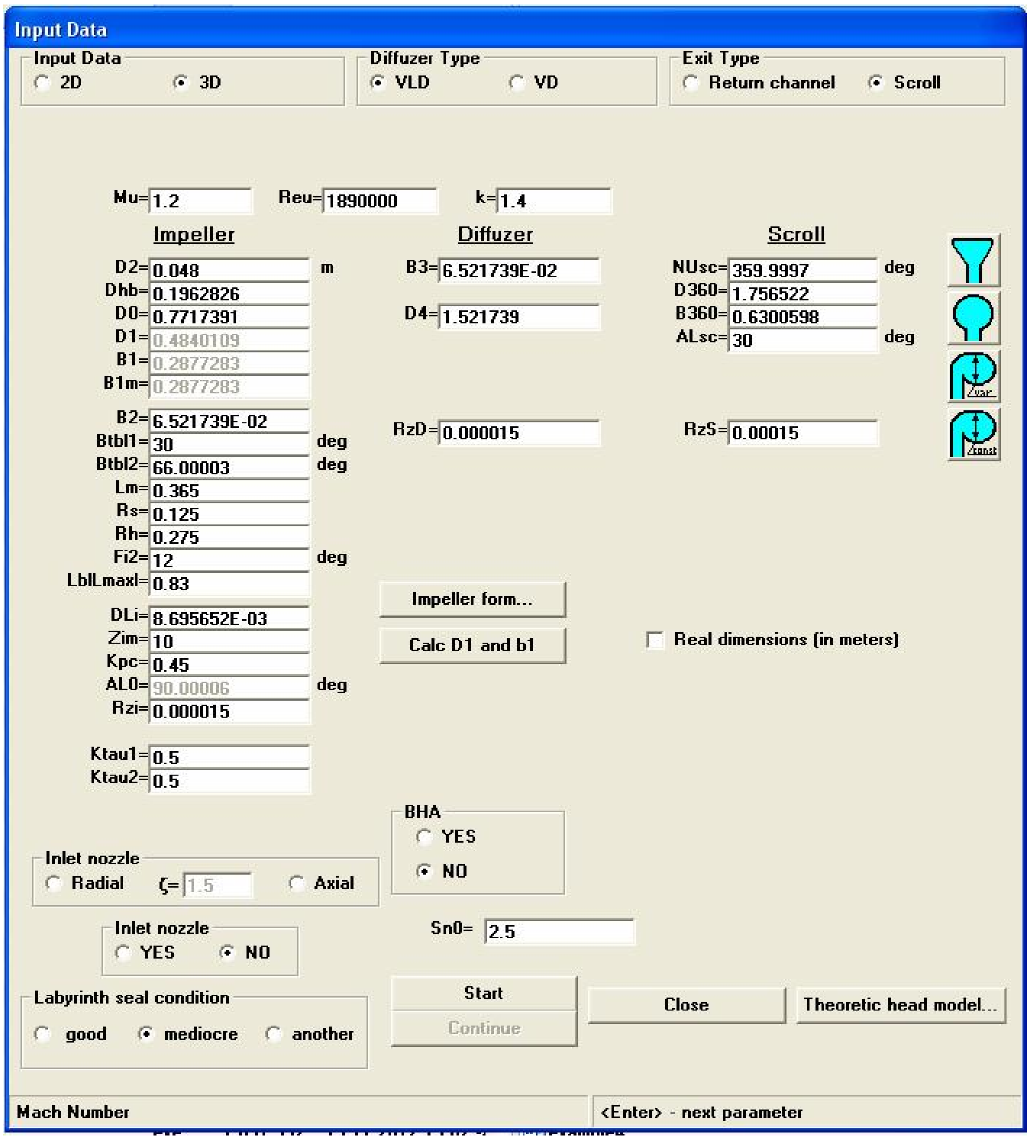

Fig.4. Input menu for a 3D impeller meridian shape (6th version of the model)

In the 7th version of the model flow parameters in 3D impellers are calculated in Q-3D mode on five blade – to – blade surfaces. The sample of flow parameters at 3D impeller exit is presented in the Fig. 5.

The new model identification and verification validated proposed improvements. Test data on TU SPb model stages 20CE family [13] were used. The accuracy of efficiency calculations at design flow rate with a single set of empirical coefficients is about 0,6% for a wide range of model stages ($\Phi_{des} = 0{,}028 \div 0{,}064$, $\psi_{T\,des} = 0{,}45 \div 0{,}68$, $\bar{D}_{hub} = 0{,}25 \div 0{,}373$, $\bar{D}_4 = 1{,}43 \div 1{,}6$ (VLD), etc.).

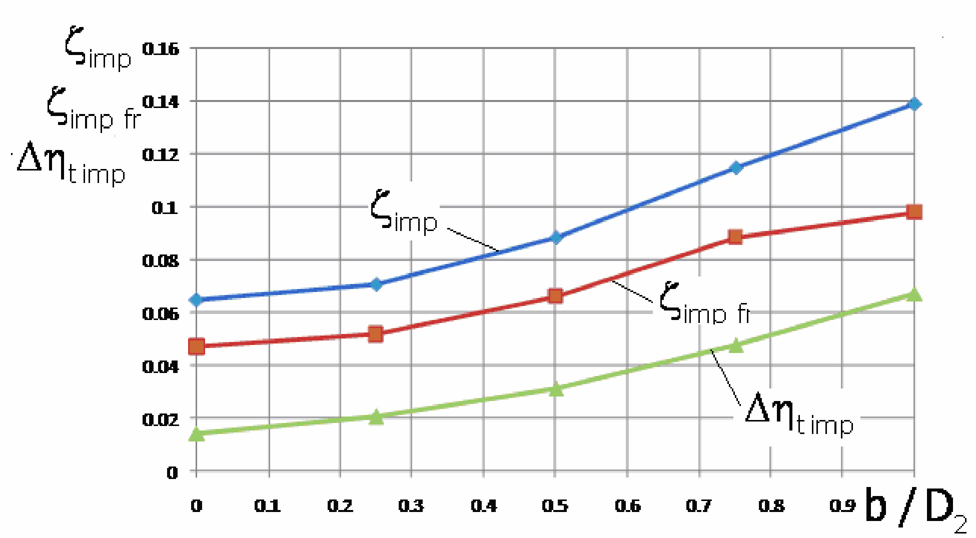

Fig. 5. Loss coefficients and loss of efficiency of a 3D impeller along blade height at an exit

Fig. 6 represents comparison of measured and calculated performances of several model stages.

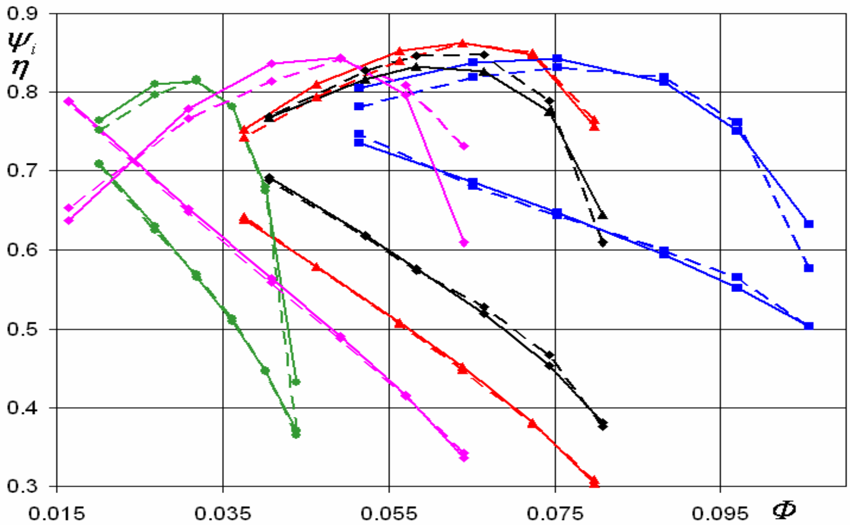

Fig. 6. Comparison of measured and calculated performances for several model stages. Dotted line - experiment, continuous line – calculation

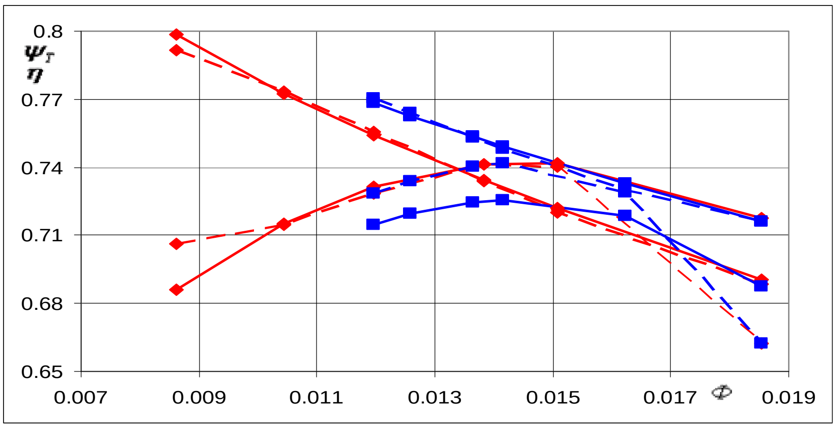

Fig. 7. Low flow rate stage measured and calculated performances. Red M_u = 0,435, blue M_u = 0,80. Dotted line - experiment, continuous line - calculation

Validation of the new model and the empirical coefficient set was made by comparison of measured and calculated performances of several low flow rate stages developed by one of compressor manufacturers. The typical result is presented in the Fig. 7. Low flow rate stage with geometry parameters $\overline{b}_2 = 0,0257$, $\overline{b}_3 = 0,0103$, $\overline{D}_4 = 1,33$, $\overline{D}_{hub} = 0,321$ was tested at M_u = 0,435 and 0,80.

The new model and 5th generation programs were applied to model plant test performances of 16 compressors (2 – 8 stages, power 4,5 - 25 MW, delivery pressure up to 12,5 MPa). Some information on results was presented also in [8, 9, 10]. Sample of modeling is presented in the Fig.8.

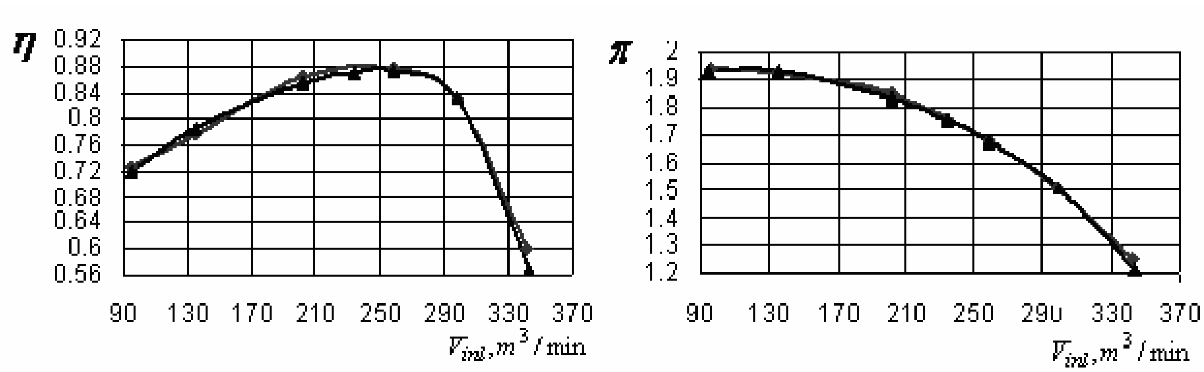

Fig. 8. Four stage 16 MWt booster compressor. Plant test performances and their modeling. ♦– test, ▲- calculation

The important result of plant test performance modeling - all performances were well correlated with calculations on the base of a single set of empirical coefficients for all 16 compressors [7, 14, 15]. The stages of the compressors can be considered as 99 model stages with range of design parameters Φ_{des} = 0,025 – 0,064, $\psi_{T\,des}$ = 0,40 – 0,85.

The new versions of the model and computer programs were also successfully applied to design 32 MWt single stage pipeline compressor, 16 MWt 6-stage booster compressor and in other designs and analytic projects (works).

NOMENCLATURE

b	height;
c	absolute velocity;
c_r	radial velocity;
c_u	tangential velocity;
c_f	skin friction coefficient;
c_w	drag force coefficient;
D	diameter;
l_{bl}	blade length;
M_u	rotational Mach number;
Re	Reynolds number;
R_{bl}	radius of blade curvature;
Rz	roughness;
$\overline{S}_{bl}$	blade area;
U	impeller periphery speed;
$\overline{V}$	volumetric flow; w – relative gas velocity;
Z	number of blades in a blade row, number of vanes;
Φ	stage flow coefficient;

η_t	total polytropic efficiency;
τ	blade blockage coefficient;
e	compressibility coefficient;
ψ_T	Euler coefficient;
ψ	stage work coefficient;
β	flow angle;
z	loss coefficient.

Subscripts

1	impeller blade inlet condition;
2	impeller tip condition;
3	vaned diffuser inlet condition;
4	diffuser exit condition;
des	design regime;
fr	friction;
inl	inlet;
max	maximum;
p	pressure side;
prof	profile;
s	suction side;
se	sudden expansion;
thr	throat;

$\bar{c} = c / u_{2,}\ \bar{b} = b / D_2$ superlinear line mean, that speed is carried to character impeller periphery speed, the linear size is carried to the character linear size (impeller periphery diameter)

REFERENCE

1. Galerkin Y., Popova E. Industrial centrifugal compressors – gas dynamic calculation and optimization concepts. Texts of the Union of the German engineers. – Aachen. – Germany. – № 1109. – 1994.
2. Galerkin Y.B., Danilov K.A., Popova E.Y. Universal Modelling for Centrifugal Compressors-Gas Dynamic Design and Optimization Consepts and Application. Yokohama International Gas Turbine Congress. – Yokohama. – 1995.
3. Galerkin Y.B., Popova E.Y., Danilov K.A., Mitrofanov V.P. Quasi-3d Calculations in Centrifugal Impeller Design. VDI Berichte. № 1425. Hannover. – 1998.
4. Galerkin Y., Danilov K., Popova E. Design philosophy for industrial centrifugal compressor. // International Conference on Compressors and their systems. – London: City University. – UK. – 1999.
5. Galerkin Y., Mitrofanov V., Geller M., Toews F. Experimental and numerical investigation of flow in industrial centrifugal impeller. // International Conference on Compressors and their systems. – London: City University. – UK. – 2001.
6. Galerkin Y. Pipeline Centrifugal Compressors – Principles of Gas Dynamic Design. International Symposium «SYMKOM-05». Compressor & Turbine Flow Systems. Theory & Application Areas. Lodz. – № 128. Vol. 1. – 2005. – P.195-209.
7. Galerkin, Y. B., Soldatova, K.V. Operational process modeling of industrial centrifugal compressors. Scientific bases, development stages, current state. Monograph. [text] // SPbTU. – 2011. (In Russian).
8. Galerkin, Y. B., Drozdov A. A., Soldatova, K.V. Centrifugal compressor efficiency types and rational application. // International Conference on Compressors and their systems. – London: City University. – UK. – 2013
9. Galerkin, Y. B., Soldatova, K.V. Universal modeling method application for development of centrifugal compressor model stages. // International Conference on Compressors and their systems. – London: City University. – UK. – 2013.
10. Seume J.R., Sextro T., Soldatova K.V. Modeling of small turbocharger compressors' performance curves. // International Conference on Compressors and their systems. – London: City University. – UK. – 2013.
11. Lojtsanskij L. Mechanics of liquid and gas. – Moscow. – 1978 (In Russian).
12. Seleznev K., Galerkin Y. Centrifugal compressors. //Leningrad. – 1982 (In Russian).
13. Galerkin, Y. B. Turbo compressors. // LTD information and publishing center. - Moscow. – 2010. (In Russian).
14. Galerkin, Y. B., Soldatova, K.V. Development of "virtual" model stages by means of 5th generation of the Universal modeling programs. [Text] // Scientific and technical journal SPbSTU. – 2011. - № 4. P. – 241-248.

15. Galerkin, Y. B., Soldatova, K.V. Development of model stages by the results of new generation industrial centrifugal compressors tests. [Text] // Proc. of 15 Intern. Compressors scientific and technical conference.— Vol. 1.— Kazan., 2011.— P. 224–232. (In Russian)