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GAS DYNAMIC DESIGNS OF CENTRIFUGAL COMPRESSORS FOR GAS INDUSTRY

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ABSTRACT

Centrifugal compressors for gas industry consume huge amount of energy. As a rule, they are single-shaft, with two or more stages and with comparatively low pressure ratio. Compressors operate at low Mach numbers and high Reynolds numbers. Two design parameters influence mostly stage performances. Stage flow coefficient optimal values lie in range 0.060 – 0.11. Chosen number of stages establishes value of this coefficient if speed of a rotor rotation is fixed. Design loading factor optimal values are 0.42 – 0.52. It corresponds to high efficiency, shifts a surge limit far from a design point and makes power maximal in a design point. Some considerations about impeller and diffuser types are presented. Design procedure consists on application of the Universal modeling programs for main dimensions optimization and performance calculations. Q3D non-viscid velocity diagrams are analyzed for optimization of blade configuration. Samples of design are presented, 32 MW single-stage pipeline compressor stage with record efficiency included.

NOMENCLATURE

$\bar{b}$	blade non-dimensional height
c_u	absolute tangential velocity (m/s)
D	diameter (m)
h_{fr}	disc friction loss of head (j/kg)
h_i	work input head of a stage (j/kg)
h_{leak}	labyrinth seal leakage loss of head (j/kg)
h_T	Euler head (head transmitted to gas by blades) (j/kg)
h_w	loss of head (j/kg)
H_i	work input head of a compressor (j)
K_n	specific speed of a compressor
K_{ns}	specific speed of a stage
M_u	rotational Mach number
$\bar{m}$	mass flow rate (kg/s)
N_T	power transmitted to gas by blades (W)
p	pressure (Pa)
R	gas constant (j/(kg*K))
Re_u	Reynolds number based on impeller diameter
T	temperature (K)
u_2	blade speed (m/s)
$\bar{V}$	volumetric flow (m^3/s)
w_1	relative velocity at an impeller inlet (m/s)
z	number of stages
$\bar{\delta}_{seal}$	non-dimensional labyrinth seal clearance
β_{fr}	disc friction coefficient
β_{leak}	labyrinth seal leakage coefficient
α_2	flow angle with respect to tangent

π	pressure ratio
φ_2	flow rate coefficient at an impeller exit
ψ_p	polytrophic work coefficient
ψ_i	work coefficient
ψ_T	loading factor
$\Delta\eta$	loss of efficiency
η	polytrophic efficiency
ρ	gas density (kg/m^3)
Φ	flow coefficient
$\Phi_{N\max}$	flow coefficient corresponding to maximum power of a stage
ζ	loss coefficient

SUBSCRIPTS

0	impeller inlet
1	impeller blade row inlet
2	impeller outlet
4	diffuser outlet
cr	surge limit
des	design
h	hub
inl	inlet
imp	impeller
max	maximum
min	minimum
st	stage
t	total thermodynamic condition

ABBREVIATION

VD	vaned diffuse
VLD	vaneless diffuse

THE PROBLEM

Centrifugal compressors are used in all basic industries. The natural gas industry is the largest consumer of centrifugal compressors. Pipeline compressors execute transportation of gas. Booster compressors rise gas pressure before a pipe when pressure becomes low in a well. Booster compressors rise gas pressure before a pipe when pressure becomes low in a well. Gas storage compressors pump gas inside storages. There are 4254 compressor units with total power 51 000 MW in Russian Gasprom only [1]. More 13 000 MW are needed in the visible future. The economic and ecological significance of energy efficiency of centrifugal compressors is evident. The first and the decisive step is its gas dynamic design.

A compressor must provide necessary mass flow rate at given delivery pressure with highest possible efficiency. Efficiency must be as high as possible along all range of a flow rate. A surge limit must be as far as possible from a design point. These are common demands for all turbo compressors. To design a new compressor flow path effectively and precisely is not easy. First Russian engineer and researcher who formulated basic principles of centrifugal compressor design was Prof. W. Ris [2]. In accordance with calculation possibilities of the time it was the set of recommendations to choose main flow path dimensions – no quantitative analysis. The authors belong to later scientific school of TU SPb that was founded by Prof. K. Seleznev [3]. The further achievements are presented in [4]. The information below is presented in accordance with TU SPb general scheme of design. Information and ideas of Western and Oriental scientists that are presented in [5], [6], [7], [8] etc. have influenced the authors' views too.

The specific features of natural gas industry compressors are connected with their parameters. Main gas transportation systems operate now at pressures 7,45 – 9,9 MPa. Undersea pipelines and gas storages operate at pressures 12 MPa and more. Pressure ratio of pipe line compressors is not high: 1,32 – 1.45. [1]. One-stage compressor can be applied in principle. But gas industry compressors have direct turbine or electric drive with limited speed of rotation. Specific speed of a compressor is established by its parameters and speed of a rotor rotation [3]. The level of efficiency is limited by specific speed of compressor stages. Pipeline compressors have two or three stages to optimize specific speed. Booster and gas storage compressors have pressure ratio 1.7 – 3.5 and up to 8 stages. Mach numbers are low in all cases. Reynolds numbers are high due to high pressures. Both factors facilitate a task to reach high efficiency. Gas turbine driven compressors must consume maximum power in a design point for better coupling with their drive.

Low pressure ratio leads to simple compressor scheme: single-shaft, equal diameters and equal loading factors of all impellers (exclusions are possible, of course):

$$D_2 = \text{const}, \psi_T = c_{u2} / u_2 = \text{const} \tag{1}$$

SPECIFIC SPEED OF COMPRESSOR AND STAGE

There are different ways to represent a compressor flow rate, head and speed of a rotor rotation by single non-dimensional coefficient. These coefficients point on main gas dynamic properties of a compressor. To analyze a compressor the authors apply the coefficient that is named a specific speed [3]:

$$K_n = 2\sqrt{\pi}\frac{\left(\bar{m} / \frac{p_{tinl}}{RT_{tinl}}\right)^{0,5}}{H_i^{0,75}} n(1/s),\ (\pi = 3.141)\cdot \tag{2}$$

Meaning condition (1) a stage of a compressor specific speed is:

$$K_{ns} = 2\sqrt{\pi}\frac{\left(\bar{m} / \rho_{tinl}\right)^{0,5}}{\left(H_i / z\right)^{0,75}} n(1/s) \approx \frac{\Phi^{0,5}}{\psi_T^{0,75}}, \tag{3}$$

where

$$\Phi = \frac{\bar{m} / \rho_{inl}}{\pi \frac{D_2^2}{4} u_2}\ (\pi = 3.141), \tag{4}$$

$$\psi_T = c_{u2} / u_2 . \tag{5}$$

Eq. (2 –5) are applied to any flow rate, and to a design point too. Coefficients and are main design parameters of a stage in accordance with [4].

FLOW COEFFICIENT Φ_{des}

A designer of a turbine driven compressor is not free to choose a design specific speed - Eq. (2). A specific speed of compressor stages depends on chosen number of stages – Eq. (3). All main dimensions and a limit of stage efficiency depend on a coefficient Φ_{des}. Detailed analysis is presented in [4]. Omitting second level factors the equation

for a relative inlet diameter that corresponds to minimal inlet velocity w_1 can be presented as:

$$\bar{D}_{1(w_{1\min})} = \sqrt{\bar{D}_h^2 + 1.26\Phi_{des}^{2/3}} \qquad (6)$$

Relative width of blades at an impeller exit is:

$$\bar{b}_2 = \frac{\Phi_{des}}{4\varphi_{2des}(\rho_2/\rho_{tinl})_{des}}, \qquad (7)$$

etc.

Flow coefficient and impeller diameters depend on number of stages that a designer decides to apply. The head transmitted to a gas in a compressor is $H_i = z\psi_i u_2^2$ if $\psi_T = \text{const}$ and $D_2 = \text{const}$. Impeller diameters of a compressor candidate with z stages related to a single-stage candidate diameter is:

$$\frac{(D_2)_z}{(D_2)_1} \approx z^{-0,5}. \qquad (8)$$

In accordance with Eq. (4) and (6) ratio of the 1st stage Φ_{des} for candidates with z (Φ_z) and with one stage (Φ_1) is:

$$\left(\frac{\Phi_z}{\Phi_1}\right)_{des} \approx \left(\frac{D_{2(1)}}{D_{2(z)}}\right)^3 \approx z^{1,5} \qquad (9)$$

The importance of Φ_{des} is connected with three main factors. The impeller total efficiency equation demonstrates it:

$$\eta_{t\,imp} = \frac{h_i - h_{r\,imp}}{h_i} = \frac{h_T + h_{fr} + h_{leak} - h_w - h_{fr} - h_{leak}}{h_T + h_{fr} + h_{leak}} = \frac{1 - 0,5\frac{\zeta_{imp}}{\psi_T}\bar{w}_1^2}{\psi_T(1 + \beta_{fr} + \beta_{leak})} \qquad (10)$$

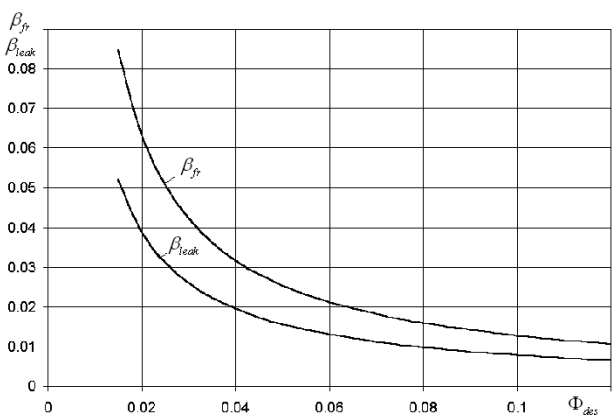


FIGURE 1. COEFFICIENTS OF PARASITIC LOSSES FOR IMPELLERS WITH DIFFERENT DESIGN FLOW COEFFICIENTS

Coefficient Φ_{des} influences all members of Eq. (8) but a loading factor. The last is one of design parameters. The authors apply their own Universal modeling method in design practice [4, 9-12]. This is the set of computer programs based on head loss models. There is the simple but effective numerical sub-program in Universal modeling to calculate parasitic losses coefficients β_{fr} and β_{leak}. The equations (9) from [3] are not quite precise but take into account main factors - Φ_{des} and $\psi_{T\,des}$:

$$\beta_{fr} = 0.012\frac{\rho_2/\rho_{t\,inl}}{\Phi \cdot \psi_T \cdot Re_u^{0.2}}, \qquad \beta_{leak} = 0.71\frac{\bar{\delta}_{seal}\bar{D}_0}{\Phi}. \qquad (11)$$

Calculations by Eq. (11) for impeller candidates with different Φ_{des} are presented in Fig. 1. It is evident that if $\Phi_{des} < 0,040 - 0,045$ a stage cannot be with high efficiency due to big parasitic losses.

The second factor is connected with flow velocity level. To bigger Φ_{des} corresponds bigger inlet diameter of an impeller – Eq. (6). The equation takes into account two main parameters - Φ_{des} and relative hub ratio $\bar{D}_h$. Graphics in Fig. 2 demonstrate influence of Φ_{des} and $\bar{D}_h$ on inlet diameter and velocity $\bar{w}_1 = \sqrt{\bar{D}_1^2 + \varphi_1^2}$.

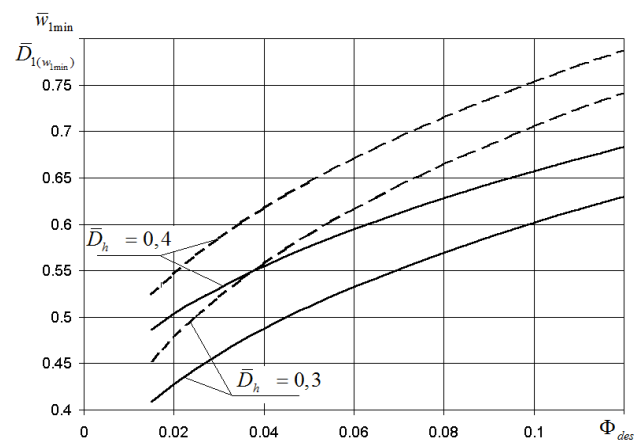


FIGURE 2. RELATIVE INLET DIAMETER AND INLET VELOCITY VERSUS DESIGN FLOW COEFFICIENT $\bar{D}_{1(w_{1\min})}$ – SOLID LINES; $\bar{w}_{1min}$ – DASH LINES

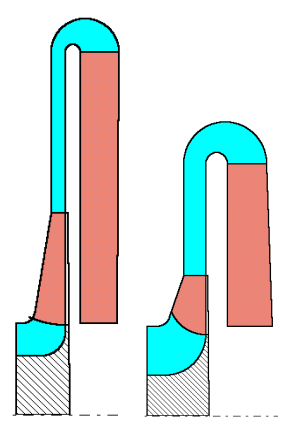


FIGURE 3. SCHEMES OF TWO STAGES WITH THE SAME DIMENSIONAL PARAMETERS AND DIFFERENT Φ_{des}. LEFT – Φ_{des} = 0.03, RIGHT – Φ_{des} = 0.09.

Impellers with larger value of Φ_{des} and $\bar{D}_h$ are inferior due to higher kinetic energy of flow as head losses are proportional to it. The opposite is influence of Φ_{des} on a loss coefficient ζ_{imp}. A scheme of two candidates of a stage with the same flow rate and pressure ratio and with different Φ_{des} is presented in Figure 3.

The less is Φ_{des} the narrower are flow path channels (Eq. (7)). Friction losses are smaller in case of bigger Φ_{des} as channels are wider. Analysis by Universal modeling loss model and design experience demonstrate that for low Mach numbers minimal sum of all head losses (i. e. maximum efficiency) is in range Φ_{des} = 0.06 – 0.075 for stages with 2D impellers, and up to 0.10 in case of 3D impellers. Efficiency drop is inevitable when $\Phi_{des} <$ 0.05. Graphic representation is in Figure 4.

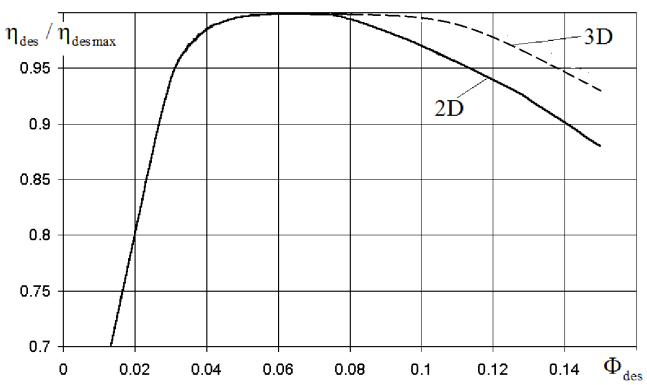


FIGURE 4. CHARACTER OF INFLUENCE OF DESIGN FLOW COEFFICIENT ON MAXIMUM EFFICIENCY OF STAGES

LOADING FACTOR $\psi_{T\,des}$

It follows formally from Eq. (10) that an impeller efficiency is higher when $\psi_{T\,des}$ is higher. Unfortunately, to higher loading factor corresponds the bigger loss coefficient ζ_{imp}. Figure 5 demonstrates that to achieve higher $\psi_{T\,des}$ larger flow deflection is necessary.

Stronger deflection leads to stronger deceleration in a blade cascade. As result, impeller efficiency is inferior for candidates with higher $\psi_{T\,des}$.

Loss of efficiency continues in a stator part of a stage when $\psi_{T\,des}$ is high due to higher kinetic energy of flow:

$$\Delta\eta_{st} = \frac{\zeta_{st}}{2\psi_T}\bar{c}_2^2 = \frac{\zeta_{st}}{2\psi_T}\left(\psi_T^2 + \varphi_2^2\right) = 0.5\zeta_{st}\left(\psi_T + \varphi_2 tg\alpha_2\right). \qquad (12)$$

Design practice and calculations by Universal modeling lead to graphic $\eta_{des} / \eta_{des\,max} = f\left(\psi_{T\,des}\right)$ shown in Fig. 6. Most effective stages tested independently or as parts of a compressor have $\psi_{T\,des}$ = 0.42 – 0.50. Such low loading factors are typical for pipeline compressors with pressure ratio 1.35 – 1.45. For booster compressor flow paths that are installed in bodies of pipeline compressors is necessary to apply higher loading factors $\psi_{T\,des}$ = 0.75 – 0.85. Two stage booster compressors have pressure ratio 1.65 – 1.70.

Loading factor $\psi_{T\,des}$ is the main parameter that influence surge limit position when Mach numbers are low [13]. Gas dynamic performances of a typical stage of 2-stage pipeline compressor are shown in Figure 7.

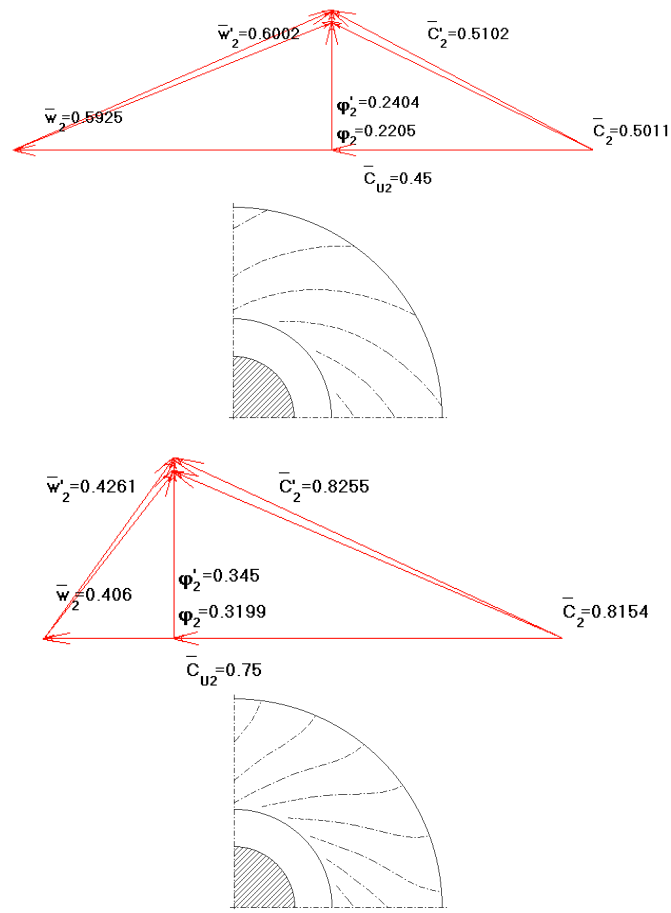


FIGURE 5. EXIT VELOCITY TRIANGLES AND BLADE CASCADE SCHEME OF IMPELLERS WITH DIFFERENT DESIGN LOADING FACTORS

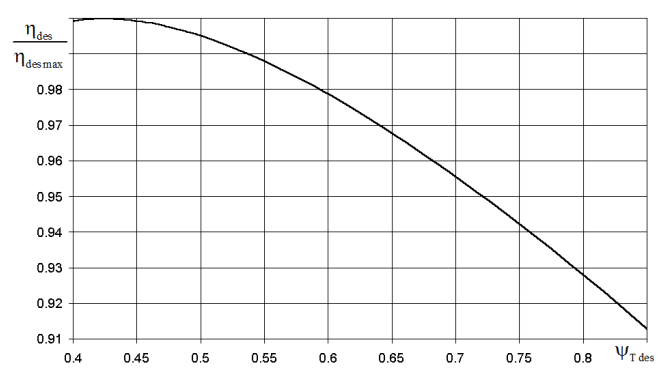


FIGURE 6. CHARACTER OF INFLUENCE OF DESIGN LOADING FACTOR COEFFICIENT ON MAXIMUM EFFICIENCY OF STAGES WITH 2D IMPELLERS

Surge limit takes place at the critical flow coefficient - Figure 7. Pressure ratio is maximal at this flow rate. It is evident that the highest pressure ratio shifts from a design point if $\psi_i = f(\Phi)$ is steep. Vast empirical data demonstrate that the function $\psi_i = f(\Phi)$ is practically linear at low Mach numbers and its steepness depends on $\psi_{T\,des}$ value. The graphic representation is presented in Figure 8.

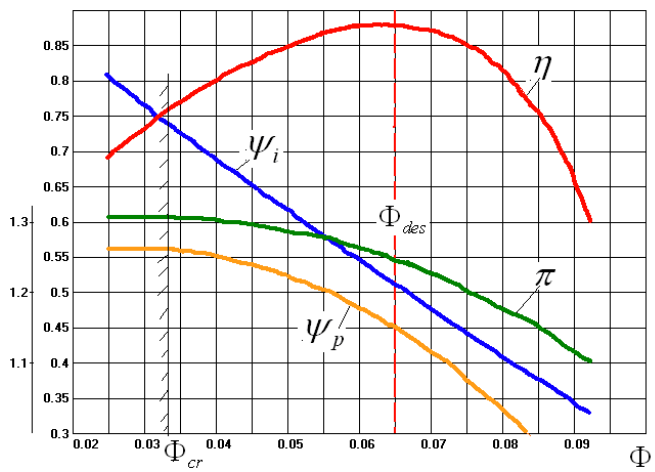


FIGURE 7. GAS DYNAMIC PERFORMANCES OF A TYPICAL STAGE OF 2-STAGE PIPELINE COMPRESSOR (OPTIMAL DESIGN BY UNIVERSAL MODELING)

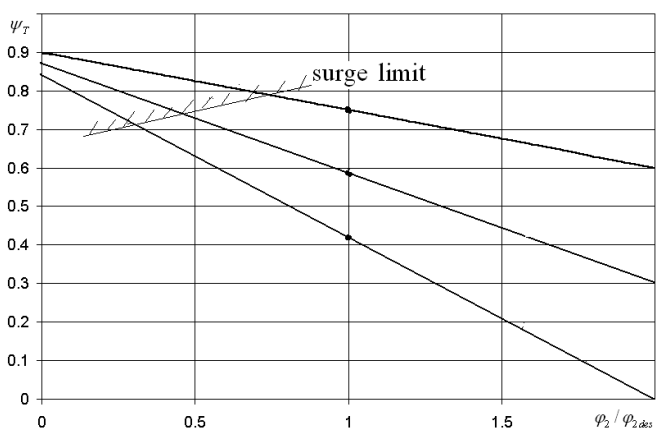


FIGURE 8. TYPICAL FUNCTIONS $\psi_T = f(\varphi_2 / \varphi_{2des})$ FOR IMPELLERS WITH DIFFERENT $\psi_{T\,des}$

The empirically proven part of $\psi_T = f(\varphi_2 / \varphi_{2des})$ lies to right from surge limit. The part to left from surge limit is an extrapolation. Surge limit coefficient Φ_{cr} / Φ_{des} is smaller if $\psi_T = f(\varphi_2 / \varphi_{2des})$ is steeper, i.e. if $\psi_{T\,des}$ is smaller.

The function $\psi_T = f(\Phi)$ is also linear practically as $\Phi = 4\bar{b}_2 \varphi_2 (\rho_2 / \rho_{t\,inl})$ and at low Mach numbers ratio $\rho_2 / \rho_{t\,inl}$ changes little. The linear function can be presented as:

$$\psi_T / \psi_{T\,des} = 1 + (\psi_{T0} / \psi_{T\,des} - 1)(1 - \Phi / \Phi_{des}), \qquad (13)$$

Polytrophic work coefficient performance can be presented as:

$$\psi_p / \psi_{p\,des} = \frac{h_T - h_w}{h_{T\,des} - h_{w\,des}} = \frac{\psi_T / \psi_{T\,des} - \dfrac{h_w}{h_{w\,des}}(1 - \eta_{des})}{\eta_{des}}. \qquad (14)$$

Model stages of pipeline compressors head loss data can be roughly approximated by the equation:

$$\frac{h_w}{h_{w\,des}} \approx 1 + 8.5 \left| 1 - \Phi / \Phi_{des} \right|^3. \qquad (15)$$

Then

$$\psi_p / \psi_{p\,des} \approx \frac{1 + (\psi_{T0} / \psi_{T\,des} - 1)(1 - \Phi / \Phi_{des}) - \left(1 + 8.5(1 - \Phi / \Phi_{des})^3\right)(1 - \eta_{des})}{\eta_{des}} \qquad (16)$$

and surge limit coefficient:

$$\Phi_{cr} / \Phi_{des} \approx 1 - \left(\frac{\psi_{T0} / \psi_{T\,des} - 1}{25.5(1 - \eta_{des})} \right)^{0.5}. \qquad (17)$$

Loading factor at zero flow rate is about $\psi_{T0} \approx$ 0.85 for stages with $\psi_{T\,des}$ = 0.45 – 0.55. The line "surge limit" in Fig.8 shows tendency.

For better matching with GT drive a compressor must consume maximum power at a design point. The power transmitted to gas by blades of an impeller is $N_T = \rho_{t\,inl} \frac{\pi}{4} D_2^2 u_2^3 \Phi \psi_T$. Power of parasitic losses is comparatively small. It is not taken into account. Using Eq. (13):

$$N_T = \rho_{t\,inl} \frac{\pi}{4} D_2^2 u_2^3 \left[\psi_{T\,des} \Phi \left(1 + \left(\frac{\psi_{T0}}{\psi_{T\,des}} - 1 \right) \left(1 - \frac{\Phi}{\Phi_{des}} \right) \right) \right] \qquad (18)$$

and:

$$\Phi_{N\max} / \Phi_{des} = \frac{\psi_{T0}}{2(\psi_{T0} - \psi_{T\,des})}. \qquad (19)$$

The ratio $\Phi_{N\max} / \Phi_{des}$ =1 takes place if:

$$\psi_{T\,des\,N\max} = 0{,}5 \psi_{T0}. \qquad (20)$$

As $\psi_{T0} \le 1$ the best matching with GT cannot be at $\psi_{T\,des} > 0.5$. Compressibility effects make compressor performances different from performances of its stages. In case of pipeline compressors these effects are hardly noticeable. Design practice demonstrates that $\Phi_{N\max} / \Phi_{des} \approx 1$ if $\psi_{T\,des}$ = 0.45 – 0.52.

TYPE OF IMPELLER

There is an opinion that 2D impellers (Figure 9, left) are inferior to 3D impellers without exceptions. The practice demonstrates that it is not true at least in case of $\Phi_{des} \le 0{,}07$, $\psi_{T\,des} \le 0{,}60$, $M_u \le 0{,}70$.

These conditions are usual for many stages of pipeline compressors. There are model stages with 2D impellers in database of R&D Laboratory "Gas Dynamics of Turbomachines" TU SPb with efficiency 88.5% for application in multistage compressors, and the stage with efficiency 90% with console disposition of the 2D impeller (more information is presented below).

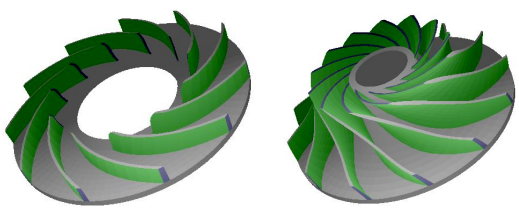


FIGURE 9. BLADE CASCADES OF 2D (LEFT) AND 3D (RIGHT) IMPELLERS (COMPUTER PROGRAM 3DM-023 FOR NON-VISCID Q3D CALCULATIONS)

TYPE OF DIFFUSER

Vaned diffusers (VD) reduce flow velocity better in given radial length $\bar{D}_4$. In stages with $\psi_{T\,des} \le$ 0.50 – 0.55 flow kinetic energy after an impeller is not very high. Vaned diffusers (VD) reduce flow velocity better in given radial length. In stages with $\Phi_{des} \le$ 0.50 – 0.55 flow kinetic energy after an impeller is not very high. Vaneless diffusers (VLD) with $D_4/D_2 \approx 1.70$ execute proper diffusion with high efficiency. The mentioned above high effective stages are provided with VLD. Performance comparison of two model stage candidates with VLD and VD is presented in Figure 10.

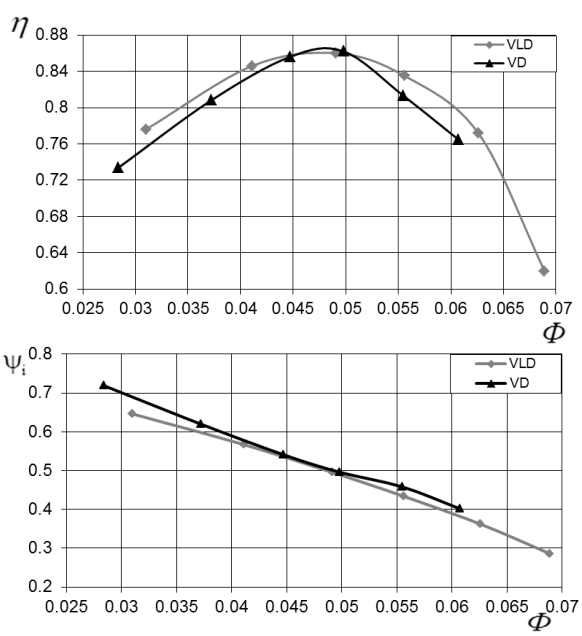


FIGURE 10. PERFORMANCES OF THE MODEL STAGE CANDIDATES WITH VLD AND VD

The diffuser radial length in given case $\bar{D}_4$ = 1.43 is not sufficient for flow diffusion in VLD. Anyway, there is no advantages of the candidate with VD nevertheless this candidate is the best among others tested with VD. Vaneless diffusers do not create problems if design loading factors are low.

The situation is different in case of an impeller with high $\psi_{T\,des}$ = 0.80. The booster compressor flow part with π = 1.70 was necessary to install in a body of a pipeline compressor (π = 1.44) with limited dimensions. It was necessary to increase loading factors and impeller diameters of the booster compressor. Flow coefficient of the first stage Φ_{des} = 0.025 is far from optimum. Application of the vaned diffuser has given a possibility to satisfy the buyer's specification. The other buyer insisted on VLD application. The performances are compared in Figure 16.

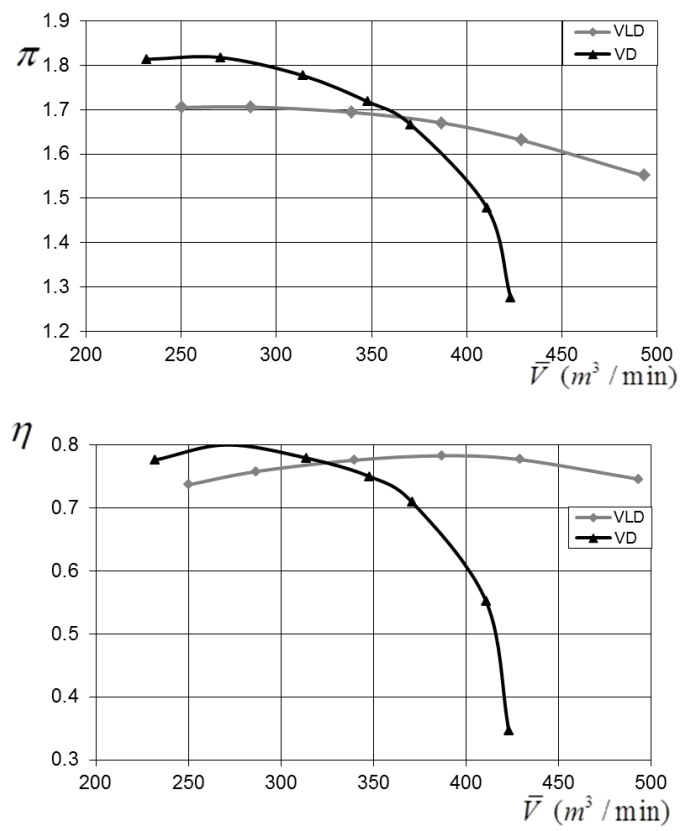
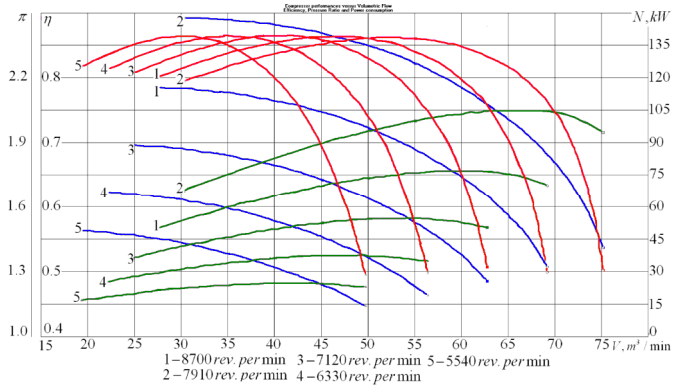


FIGURE 11. PERFORMANCES OF TWO BOOSTER COMPRESSORS WITH VD AND VLD

The candidate with VD is more effective at small flow rates. Its surge limit is not inferior.

DESIGN PRACTICE FEATURES AND SAMPLES

The heart of optimal design computer programs are math models – the system of algebraic equations with 65 empirical coefficients [10], [11], [12]. The programs calculate gas parameters in control sections of a flow path and performance maps of stages and compressors. Menu and samples of information are presented in Figure 12. The program determines main dimensions of a flow path.



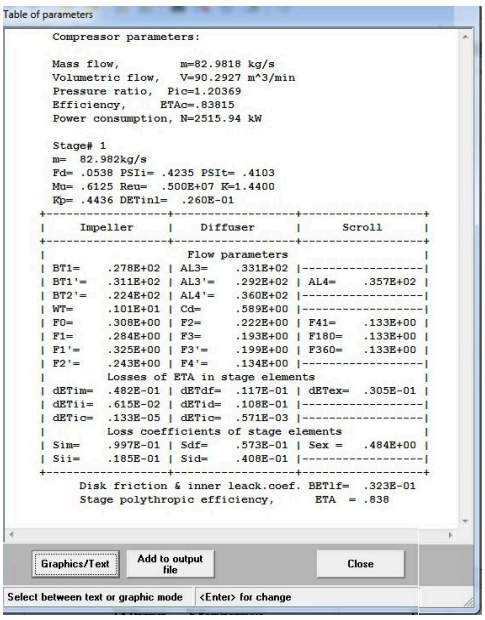


FIGURE 12. SAMPLES OF INFORMATION OF COMPUTER PROGRAM FOR A COMPRESSOR PERFORMANCE MAP CALCULATION

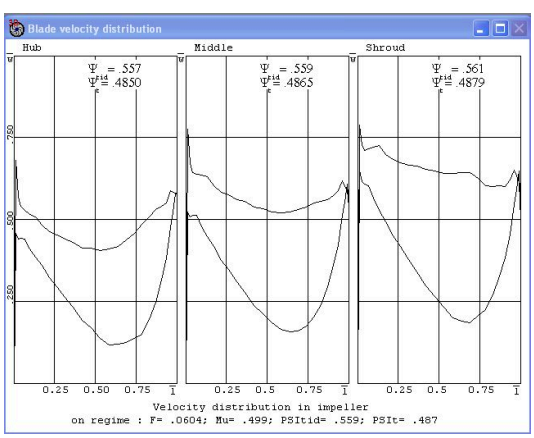


FIGURE 13. SAMPLE OF A VELOCITY DIAGRAM WITH CONTROLLED FLOW DIFFUSION ALONG SUCTION SURFACE. THREE BLADE-TO-BLADE SURFACES

Blade geometry is established on a base of Q3D non-viscid velocity diagram analysis. The program 3DM-023 is a combination of quasi-orthogonal and integral equations methods. The sample of a velocity diagram with controlled flow diffusion along suction surface is presented in Figure 13.

The authors are not sure in validity of performance calculations by CFD programs and prefer Universal modeling programs – sample is above in Fig. 12. The validation of the Universal modeling is supported by design practice since mid-1990th. More than 400 compressors with power 1.5 – 25 MW are constructed by the Russia's and Ukraine's manufacturers. The booster compressor (mentioned in Part 8) scheme and performances are presented in Figure 14.

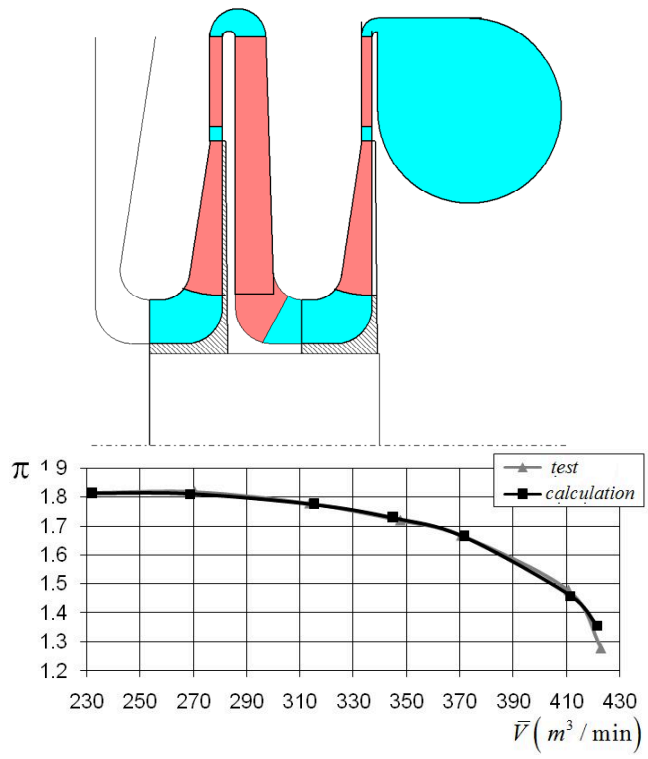


FIGURE 14. THE BOOSTER COMPRESSOR SCHEME AND PERFORMANCES. SIMULATION BY 6TH VERSION OF THE UNIVERSAL MODELING

The efficiency and pressure ratio performances are simulated by new versions of the programs [10], [11] with good precision. In other cases simulation was less impressive but inaccuracy seldom exceeds 0,5%.

The high-effective single – stage 32 MW pipeline compressor that was designed by the authors for a Ukrainian partner [14]. Scale 1:2 model test was made by the partner. The model cross-section view is presented in Figure 15.

The partner offered the effective general scheme – single-stage with axial inlet nozzle and unlimited radial dimension. High-speed drive application has made possible single-stage scheme.

Test results, calculated design performances and CFD-simulation are presented in Figure 16.

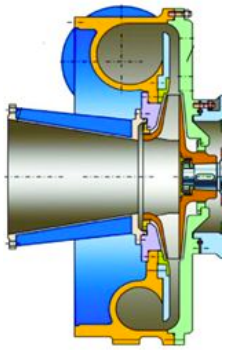


FIGURE 15. SCALE 1:2 MODEL OF THE 32 MW PIPELINE COMPRESSOR

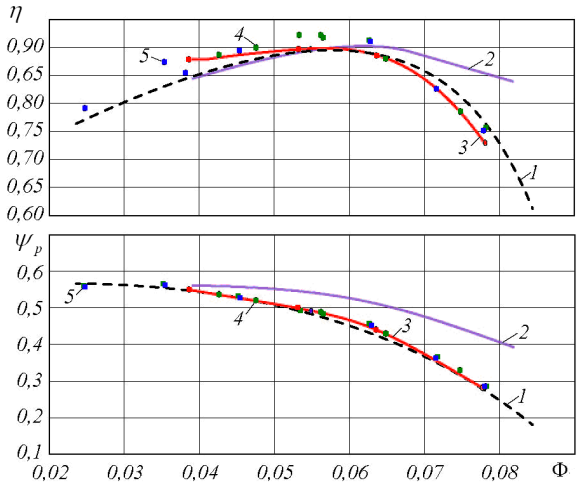


FIGURE 16. COMPRESSOR 32 MW MODEL CALCULATED PERFORMANCES AND TEST RESULTS. 1-5TH VERSION UNIVERSAL MODELING DESIGN PERFORMANCES, 2- ANSYS CFX. TESTS: 3- M_u = 0,705, 4- M_u = 0,710, 5- M_u = 0,700

Tests were made three times with close rotation frequency and corresponding Mach numbers. Design performance prediction is the most satisfactory in all authors' practice.

The maximum efficiency 90% is reached due to proper optimal design and favorable constructive principles. Console impeller is preceded with an axial inlet nozzle. Zero hub ratio diminished the impeller inlet diameter and inlet velocity. The vaneless diffuser has an optimal length. Hardly this level of efficiency can be reached by multistage compressor. Zero nub ratio and an axial inlet nozzle are impossible in multistage compressors.

CFD-simulation of the design was made by the industrial partner. These calculated performances are shifted to bigger flow rates while maximum efficiency is predicted well – curves 2 in Figure 16. The same results are presented in [15]. The authors own calculations have demonstrated similar effect. Precise simulation of centrifugal compressor performances by CFD-methods is spill uncertain.

CONCLUSION

The presented ideas in many cases are related to all centrifugal compressors as well. But the difference must be taken into account. Stages with bigger loading factors and operating at high Mach numbers must be treated the other way. The authors are aware of the fact that most manufacturers know presented above considerations. The authors are ready to discuss their opinions with the interested specialists who have other points of view.

These considerations may be of interest to compressor final users. Discussing a new compressor design users sometimes insist on compressor features that do not correspond to compressor parameters.

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