

SAMPLE OF CFD OPTIMIZATION OF A CENTRIFUGAL COMPRESSOR STAGE

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Abstract: Industrial centrifugal compressor stage is a complicated object for gas dynamic design when the goal is to achieve maximum efficiency. The Authors analyzed results of CFD performance modeling (NUMECA Fine Turbo calculations). Performance prediction in a whole was modest or poor in all known cases. Maximum efficiency prediction was quite satisfactory to the contrary. Flow structure in stator elements was in a good agreement with known data.

The intermediate type stage “3D impeller + vaneless diffuser+ return channel” was designed with principles well proven for stages with 2D impellers. CFD calculations of vaneless diffuser candidates demonstrated flow separation in VLD with constant width. The candidate with symmetrically tapered inlet part $b_3 / b_2 = 0,73$ appeared to be the best. Flow separation takes place in the crossover with standard configuration. The alternative variant was developed and numerically tested. The obtained experience was formulated as corrected design recommendations.

Several candidates of the impeller were compared by maximum efficiency of the stage. The variant with gas dynamic standard principles of blade cascade design appeared to be the best. Quasi – 3D non-viscid calculations were applied to optimize blade velocity diagrams – non-incidence inlet, control of the diffusion factor and of average blade load. “Geometric” principle of blade formation with linear change of blade angles along its length appeared to be less effective.

Candidates’ with different geometry parameters were designed by 6th math model version and compared. The candidate with optimal parameters - number of blades, inlet diameter and leading edge meridian position – is 1% more effective than the stage of the initial design.

Nomenclature

$\bar{b}$ - blade relative width;

c_m – meridian velocity, m/s;

D - diameter, m;

$\bar{D}_h$ - hub relative diameter;

$\bar{D}_0$ - impeller inlet relative diameter;

$\bar{D}_{impr}$ - average relative tapering part diameter;

k - Isentropic coefficient;

$\bar{l}_m$ - relative length in meridian plane;

$\bar{L}_m$ - relative axial length of impeller;

M - Mach number;

$$M_u = \frac{u_2}{\sqrt{kRT_{inl tot}}};$$

M_z - blade load momentum;

r, R - radius;

Re - Reynolds number;



$$\text{Re}_u = \frac{u_2 D_2}{\mu_{inl}} \frac{P_{inl,tot}}{RT_{inl,tot}};$$

u_2 - impeller periphery speed, m/s;

$\bar{w} = \frac{w}{u_2}$ - relative velocity;

w' - velocity in view of a blade blockade factor, m/s;

z - number of blades;

α_2 - flow angle at an outlet of the impeller (to tangential direction), $^\circ$;

β_{bl} - blade angle to tangential direction, $^\circ$;

ρ - gas density, kg/m³;

ζ - loss coefficient;

ζ_{fr} - friction loss coefficient;

ζ_{mix} - mixing loss coefficient;

ε'_1 - densities' ratio at inlets of impeller and of an impeller;

τ - blade blockade factor;

$\psi = \frac{h_p}{u_2^2}$ - polytrophic head coefficient;

$\psi_T = c_{u2} / u_2$ - Euler work coefficient;

η^* - total efficiency;

Φ - flow rate coefficient;

Subscripts

0 – impeller inlet;

1 – impeller blade row inlet

2 – impeller outlet

3 - diffuser inlet

4 – return channel inlet;

des – design;

imp – impeller;

min – minimum;

p – pressure;

s – suction;

VLD – vaneless diffuser

1. SCOPE OF THE WORK

High specific speed stages have smaller radial dimensions. A high specific speed stage guarantees higher efficiency as the 1st stage of a multistage industrial compressor. If design flow rate coefficient Φ_{des} exceeds 0.07 – 0.075 well studied 2D impellers are not effective. Less studied and more complicated in flow path configuration 3D impellers with axial - radial blade disposition must be applied. Stator elements of high flow rate stages are wider that leads to specific flow behavior.

Industrial compressor stages with 2D impellers were objects of attention of the elder of authors for decades. Theoretic background, design principles and positive design practice are presented in [1]. 3D impellers have much more complex configuration. It is difficult to formalize and much more experiments are necessary to formulate design recommendations or obtain optimal solution of an exact impeller. CFD numerical experiments can be an effective instrument for gaining design experience. Universal modeling method – complex of computer programs for centrifugal compressors and stages performance curves calculations [2, 3, 4, 5, 6, 7] can be applied too.

The object of the optimization experiments is a stage “3D impeller + vaneless diffuser + return channel (crossover included)”. The design parameters are $\Phi_{des} = 0.105$, $\psi_{Tdes} = 0.56$, $M_u = 0.7$, $\text{Re}_u =$

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 5600000, $k = 1.4$. Design flow rate, Euler work coefficients and Mach number correspond to demand of modern pipeline compressors and alike. The similarity criteria Re_u and k correspond to air tests of a model stage. The design was made by means of Universal modeling method that is based on general principles formulated in [1] and that was presented at several conferences [3, 5, 6, 7].

The CFD numerical experiment was made by means of NUMECA Fine Turbo. The authors experience and publications [8, 9, 10, 11, 12], lead to the next conclusions about CFD calculations results:

- efficiency performance curve calculation for stages with radial impellers is more or less satisfactory;
- work coefficient prediction is overestimated on 6 - 7% at design flow rate for 2D and 3D impellers;
- 3D impeller calculated performance curves are shifted right;
- surge limit prediction of 3D impellers is unsatisfactory. Calculated surge limit is very close to the regime of maximum efficiency;
- calculated maximum efficiency of a stage with 3D impeller is very close to the measured value;
- loss coefficients of stator parts calculated by engineering Universal modeling method and CFD calculated are very close.

The authors keep in mind these considerations while planning and interpreting numerical experiments.

2. ANALYSIS OF STATOR STAGE ELEMENTS

Figure 1 demonstrates meridian configuration of the initial design and streamlines calculated by NUMECA Fine Turbo – design flow rate.

Absolute velocity angle at the VLD exit is 26.7° . The recommendation for VLD with constant width of medium flow rate stages is $\alpha_{2des} \geq 25^\circ$ [1]. However CFD calculation points on flow separation at the very beginning of VLD at Figure1. The separation continues in the crossover. ANSYS CFX calculations of VLD with uniform flow at inlet and without crossover are presented in [11, 12]. For the VLD of the same configuration flow separation is expected if $\alpha_2 < 20 - 22^\circ$. The negative influence of flow non-uniformity at the impeller exit is evident.

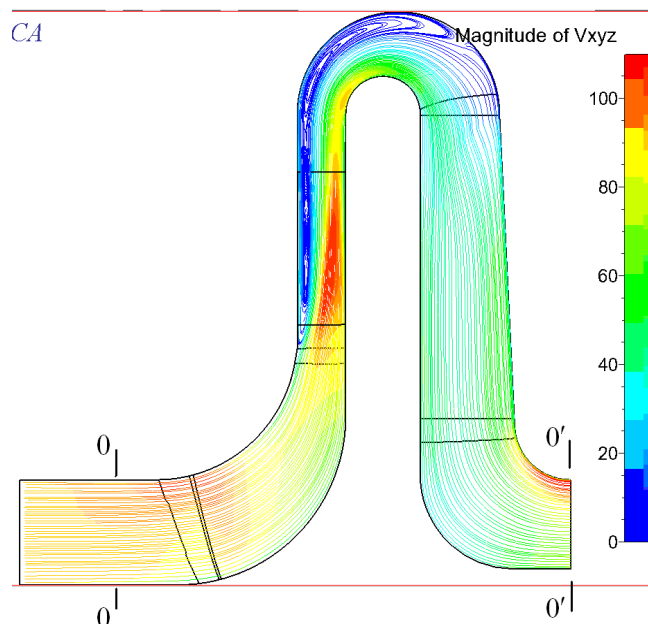


Figure 1. Initial design meridian configuration and streamlines (design flow rate, NUMECA Fine Turbo)

Tampering of VLD initial part is the known way to avoid flow separation for a long time [13].

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 Five candidates "impeller + VLD" were designed and compared. Geometry parameters of them are presented in the Table 1. The 1st candidate is the initial design. The others have initial parts with different tapering. The tapering is made from the hub side in three cases. This kind of tapering is recommended in [13] as effective gas dynamic solution. It helps also to diminish a stage axial length.

The candidates ## 4, 5 have longer tapering part. The candidate # 5 has symmetrically tapered initial part.

Table 1
 Geometric parameters of VLD candidates

#	1	2	3	4	5
Tapering side	-	hub	hub	hub	symmetric
$\bar{b}_3$	0.0743	0.0628	0.0543	0.0543	0.0543
b_3 / b_2	1	0.845	0.731	0.731	0.731
$\bar{D}_{tpr}$	-	1.16	1.16	1.32	1.32

The candidates' ## 1, 2, 3 configuration and streamlines are shown at Figure 2.

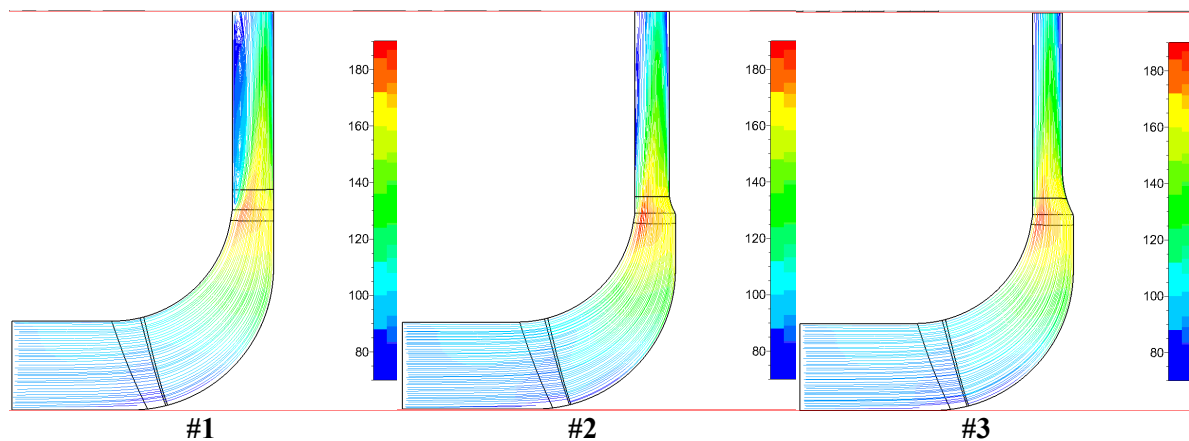


Figure 2. VLD candidates meridional configuration and streamlines (design flow rate, NUMECA Fine Turbo)

The flow separation in initial design is lesser than in the complete stage at Figure 1. Negative influence of the crossover is evident. Tapering ratio $b_3 / b_2 = 0.845$ suppressed separation but has not eliminated it completely. The further tapering to $b_3 / b_2 = 0.731$ eliminated flow separation. This tapering was accepted for further candidates.

The recommendation of [15] in tapering configuration was proven by calculation of two more candidates ## 4, 5. Candidate #4 has twice longer initial tapering part $\bar{D}_{tpr} - 1.132$. Candidate #5 has initial tapering part $\bar{D}_{tpr} - 1.132$ but the tapering was symmetric, i.e. was made from shroud and hub sides. The stages with different tapering were calculated by NUMECA Fine Turbo. The influence of tapering mode was noticeable. The comparison of three VLD loss coefficient performance curves is presented at Figure 3.

Symmetrical tapering (VLD #5) reduces the loss coefficient approximately to 10% in comparison with hub steep tapering (VLD #3). Rise of the stage efficiency is 0.4%.

To diminish negative influence of the crossover on a diffuser its configuration was modified. The initial design hub and shroud surfaces are formatted by arcs as it is shown at Figure 1. The modified configuration is presented at Figure 4 (right). The shroud surface is formatted by two arcs and a straight line. Streamlines are presented too (design flow rate, NUMECA Fine Turbo). The VLD tapering ratio is $b_3 / b_2 = 0.731$ so there is no flow separation in this element. But there is the

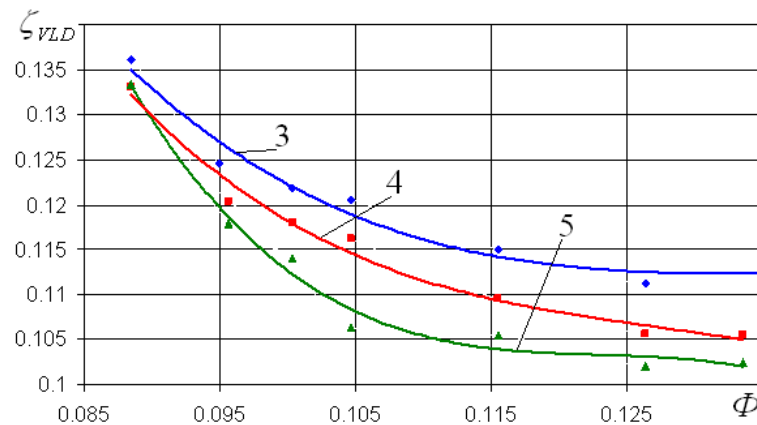


Figure 3. Loss coefficient performances of vaneless diffusers with different tampering

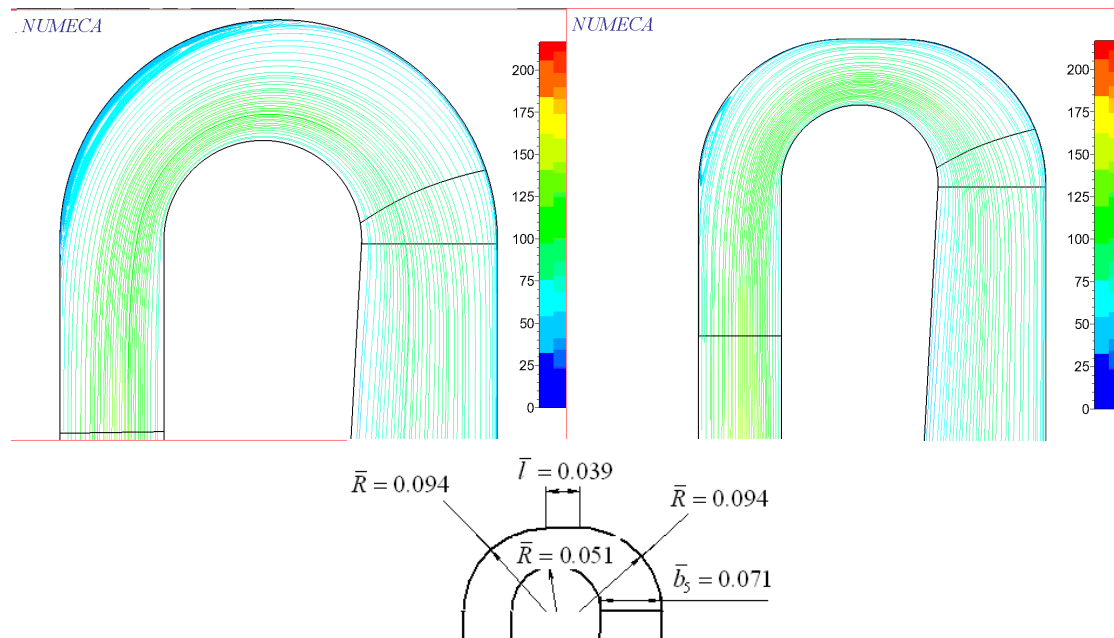


Figure 4. Left and center – initial and modified shroud surface and streamlines.
Right – modified shroud surface configuration

3. 3D IMPELLER BLADE CASCADE PROFILING AND OPTIMIZATION

Meridian dimensions and hub/shroud shape of the initial design were chosen as recommended in [1]. Blade cascade profiling is based on the choice of functions $\beta_{bl} = f(l_{mi}/l_m)$ on three blade to blade surfaces: shroud, middle, hub. An expert assessment of non-viscid Q3D velocity diagrams is applied. Figure 5 (left) demonstrates initial meridian configuration.

Four blade cascade candidates were designed. The 1st candidate is designed by the “geometry” principle that is known but is not applied in TU SPb practice. Three linear functions $\beta_{bl} = f(l_{mi}/l_m)$ of this candidate are shown at Figure 5. Inlet blade angles at three surfaces were chosen to minimize incidence losses along a leading edge height. The outlet angle was chosen in accordance with given Euler work coefficient. The chosen number of blades establishes an average blade load.

The TU SPb principle of profiling is realized in the candidate Imp-2. Function $\beta_{bl} = f(l_{mi}/l_m)$ at a shroud blade-to-blade surface has maximum. It helps to control velocity gradient along suction side of blades. There is an opinion that increase of the blade outlet angle from hub to shroud diminishes flow non-uniformity in the meridian plane.

The principle of the blade outlet angle increasing from hub to shroud is not proven so the candidate Imp-3 differs from Imp-2 by the constant outlet blade angle – Figure 5.

The principles of the candidate #4 is the same as for #2 but its relative length $\bar{L}_m$ is 0.35. The relative length of candidates ## 1 -3 is 0.30. View of blade cascades is shown on Figure 6. Velocity diagrams on hub and shroud surfaces for all candidates are shown on Figure 7.

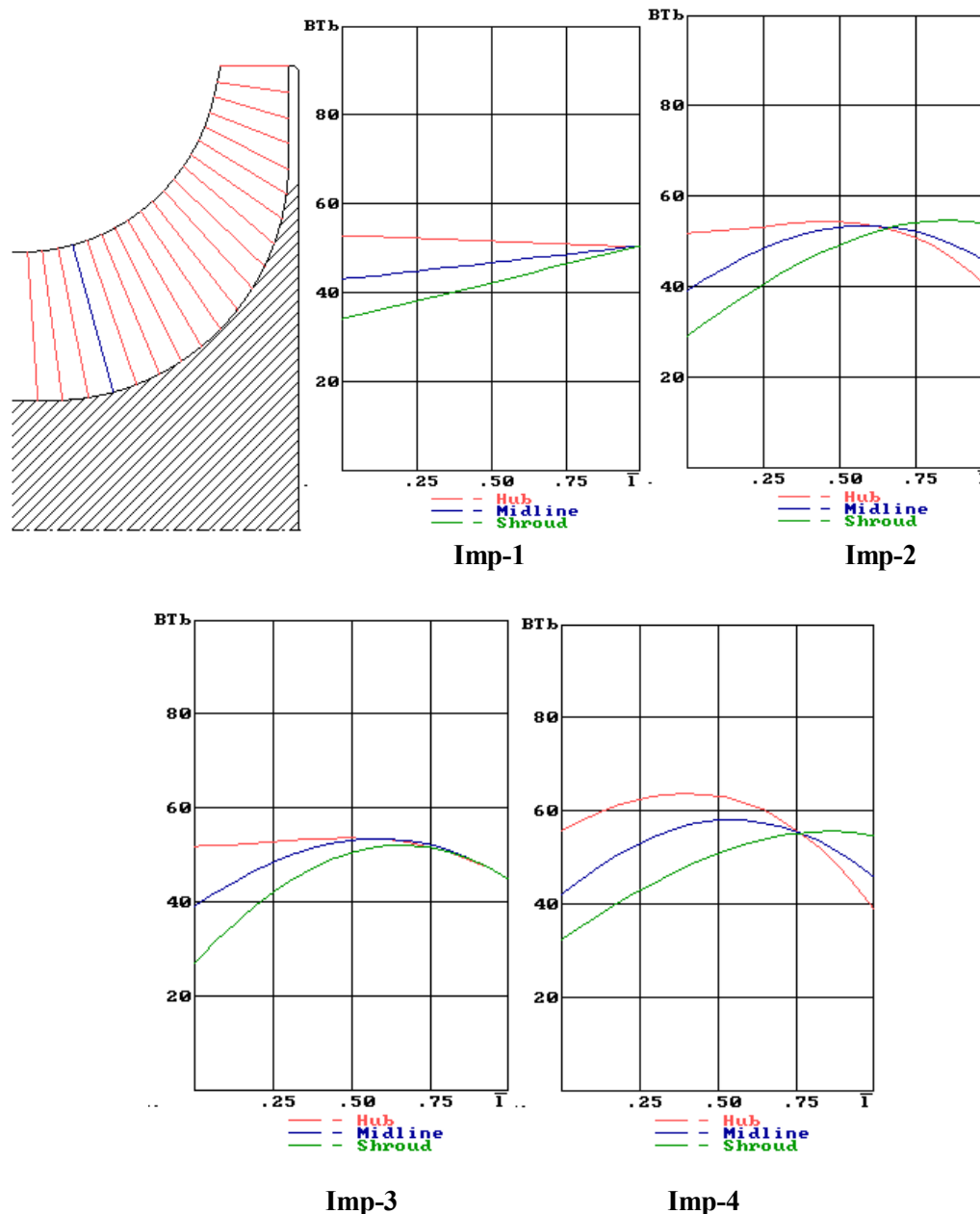


Figure 5. Meridian configuration of the impeller and the blade angles on three blade to blade surfaces – four candidates

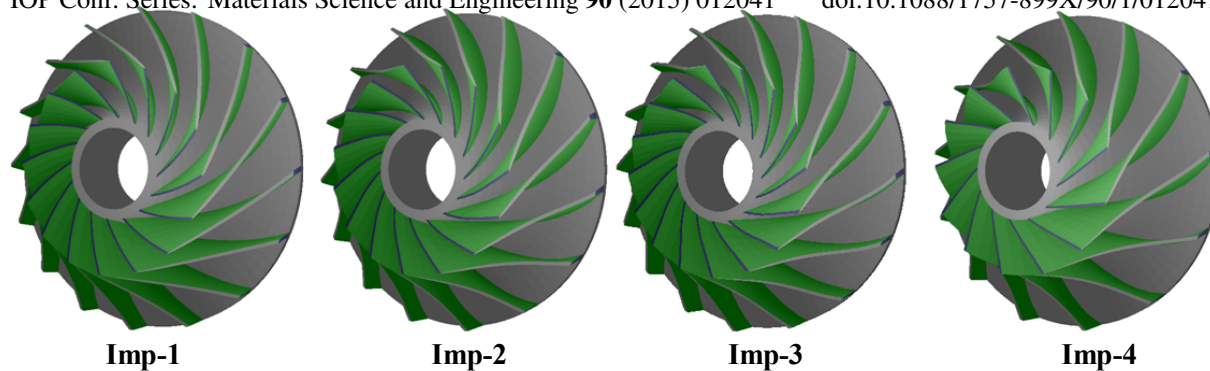


Figure 6. View of blade cascades – four candidates

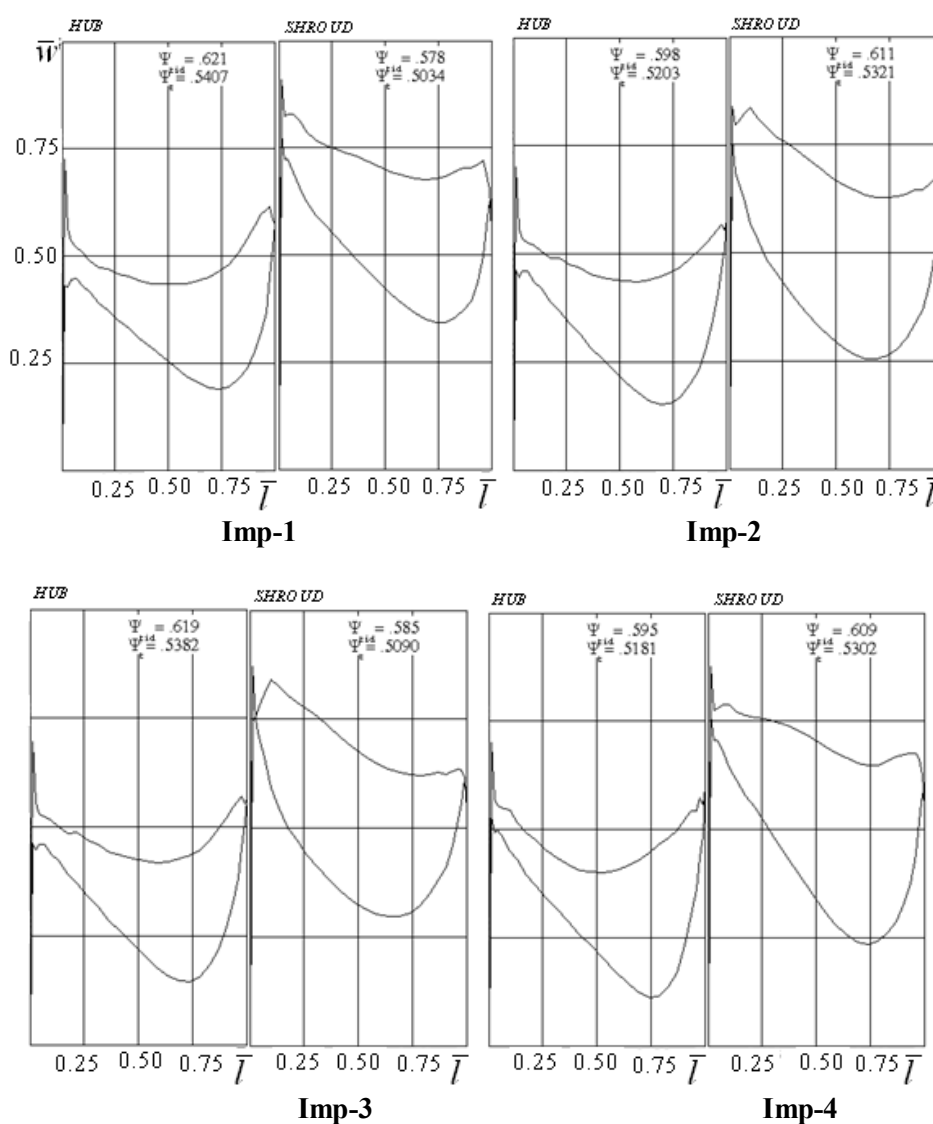


Figure 7. Velocity diagram on hub and shroud surfaces – four candidates

Calculations of performance curves of stages with four impellers – candidates lead to the next conclusions. In accordance with Q3D design program $\psi_{T_{des}} = 0.56$ of all four impellers – candidates.

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 CFD calculation predicts $\psi_{Tdes} \approx 0.65$ for Imp-1, 2, 4 and 0.66 for Imp – 3. Mismatch with experiments of CFD calculation is a usual event. But the mismatch value 12-18% is too big and the problem deserves additional study.

Total efficiency performance curves – Eq. (1), (2) of the stages and impellers are presented at Figure 8.

$$\eta^* = \frac{\ln \frac{p_{0'}^*}{p_0^*} \frac{k-1}{k}}{\ln \frac{T_{0'}^*}{T_0^*}} ; \quad (1)$$

$$\eta_{imp}^* = \frac{\ln \frac{p_2^*}{p_0^*} \frac{k-1}{k}}{\ln \frac{T_2^*}{T_0^*}} . \quad (2)$$

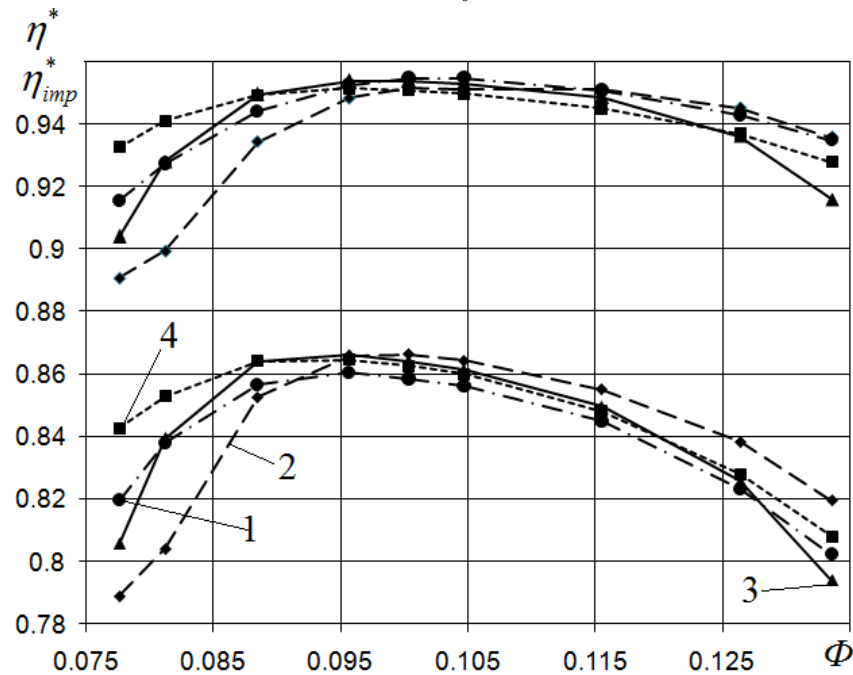


Figure 8. Total polytropic efficiency of impellers and stages

It is well-known fact that an impeller total efficiency is not an objective parameter unlike stage efficiency. In the experiments and in CFD calculations averaged total pressure at an impeller outlet – section “2” - does not reflect impeller losses. Impeller’s mixing losses do not occur in impeller, but in a following diffuser. The most effective Imp-3 demonstrated the most uniform outlet flow in the meridian plane. Imp-2 has less uniform outlet flow (the idea to diminish non-uniformity by variable outlet angle was not realized in this candidate). In spite of it the most effective is the stage with Imp-2 designed by principles well proven for 2D impellers [1].

The less effective is the stage with Imp-1 – “geometry” design.

The Imp-2 ($\bar{L}_m = 0.30$) and #4 ($\bar{L}_m = 0.30$) are designed on similar principles. Maximum efficiency of the stages differs little. Performance curves of the stage with Imp-4 shifted to left in comparison with the stage with Imp-2.

4. 3D IMPELLER OPTIMIZATION BY 6TH UNIVERSAL MODELIND VERSION

While CFD calculation is a proper mean to study details of a flow path configuration, the main simple geometry parameters can be studied by engineering calculation methods. The 6th math model and the proper version computer programs were applied for the optimization of three geometry parameters of the stage. All compared stage's candidates were designed by the standard method of the R&D Laboratory "Gas dynamics of turbo machines". The scheme at Figure 9 shows meridian dimensions of 3D impeller.

The principle of optimization is simple. The efficiency loss depends on inlet flow kinetic energy and loss coefficient:

$$\Delta\eta_{imp} = \frac{\zeta_{imp}}{\psi_T} 0,5 \left(\frac{w'_{1s}}{u_2} \right)^2 = \frac{\zeta_{imp}}{\psi_T} 0,5 \left(\frac{c_{m1}'^2 + u_{1s}^2}{u_2^2} \right). \quad (3)$$

Euler work coefficient is represented in this equation but it is given for the stage. To minimize efficiency loss is necessary to minimize loss coefficient and inlet velocity. The standard procedure for an impeller inlet diameter choice is based on the principle of an inlet relative velocity minimization.

The more is an inlet diameter the more is u_{1s} and less c_{m1}' . The minimum of the relative velocity corresponds to an inlet diameter by the equation that is known from 1930th:

$$\bar{D}_{0wmin} = \sqrt{\bar{D}_{hub}^2 + 2^{\frac{1}{3}} \left(\frac{\Phi_{des}}{\varepsilon'_1 \tau_1} \right)^{\frac{2}{3}}}. \quad (4)$$

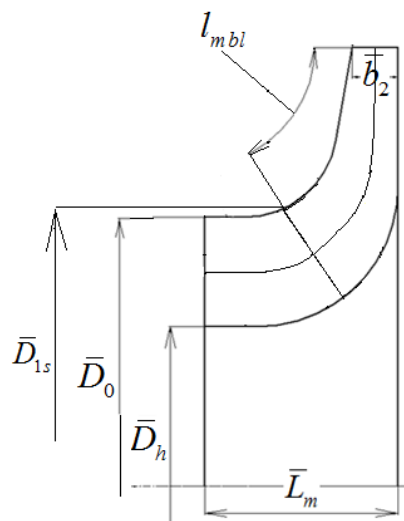


Figure 9. Meridian dimensions of 3D impeller

Different ζ_{imp} values correspond to different inlet diameters $\bar{D}_0$. The optimal inlet diameter $\bar{D}_0$ can be equal, less or more than $\bar{D}_{0wmin}$. The coefficient $A_d = \bar{D}_0 / \bar{D}_{0wmin}$ value is used in design practice as design recommendation based on experiments – physical or numerical.

The design experience has demonstrated that the loss coefficient of 2D impellers depends little of $\bar{D}_0$. The coefficient A_{dopt} is equal to 1 in most cases.

The choice of $\bar{D}_0$ is not so simple in case of 3D impeller but firstly influence of blade number was investigated.

The initial stage for the blade number optimization has relative inlet diameter $\bar{D}_0 = 0.60$, and the relative leading edge position $\bar{l}_{mbl} = 0.792$. The result of blade number influence is presented in the Table 2.

Table 2

Stage's candidates with different blade number

z	$\beta_{bl1}, ^\circ$	$\beta_{bl2}, ^\circ$	w'_{1s} / u_2	ζ_{fr}	ζ_{mix}	$\eta, \%$
10	27.9	67.9	0.717	0.0597	0.0335	84.8
12	28.7	57.9	0.717	0.0674	0.0225	85.2
15	29.7	51.1	0.719	0.0801	0.0127	85.3
18	30.5	48	0.721	0.0930	0.0073	85
22	31.4	46.2	0.724	0.1100	0.0043	84.5

The number of blades was varied in the wide range of $z = 10 - 22$. The gas dynamic flow parameters $\Phi_{des} = 0.105$, $\psi_{Tdes} = 0.56$ are given for all candidates. Therefore all impellers – candidates have different blade angles. The less is a blade number – the bigger is a blade outlet angle to obtain given ψ_{Tdes} . Inlet blade angles are bigger for candidates with bigger blade number due to blade blockade factor.

The maximum efficiency corresponds to 15 blades. The bigger blade load corresponding to smaller blade number leads to earlier flow separation and bigger mixing losses. Big blade numbers increases blades' surface area and friction losses increase as consequence.

Math model calculations demonstrate the next quantitative results. The candidate with 22 blades has friction losses on 1.84 times more in comparison with the candidate with 10 blades. Mixing loss coefficient 0.00425 is very little if $z = 22$. If $z = 10$, the separation is sufficient and mixing loss coefficient is 0.0335 that is comparable with friction loss coefficient. Optimal number - fifteen blades well correlates with the design procedure recommendations.

The inlet diameter by eq. (4) for the analyzed impeller with 15 blades is equal to 0.613. The Eq. (4) estimates inlet relative velocity correctly if the leading edge is disposed at the section "0". Its position for an impellers analyzed below is determined by $\bar{l}_{mbl} = 0.792$. The difference of $\bar{D}_0$ and $\bar{D}_{1s}$ is insignificant so the calculation results in the Table 3 are correct practically.

Candidates with $A_d < 1$ are more effective. The optimal value of $\bar{D}_0$ lies within 0.55 - 0.56, $A_{dopt} = 0.897 - 0.914$. The stage with optimum $\bar{D}_0$ has efficiency 0,861. The stage with $A_d = 1.0$ is 1.3 % less effective. The less is $\bar{D}_0$ if $A_d < 1$ the higher is kinetic energy at an impeller inlet. The efficiency becomes higher in spite of it. The reason is that blade surfaces are smaller. That diminishes friction loss coefficient in spite of higher average velocity w'_{1s} / u_2 that influence friction losses too. The positive factor is that flow deflection is less in impellers with smaller $\bar{D}_0$. It is important to note that the outlet blade angle of all candidates is the same, $\beta_{bl2} = 51^\circ$. The momentum that creates Euler

work and is created by blade load is: $M_z = z \sum_0^2 \rho \cdot w \cdot (w_s - w_p) \cdot b \cdot dl_m \cdot r$. The more is $\bar{D}_0$ the

more is a blade height b and less is an average flow velocity w . But that does not mean that their product $w \cdot b$ is the same for all candidates. The load $w_s - w_p$ depends on flow deflection, i.e.

depends on blade angle gradient $\partial \beta_{bl} / \partial l_m \sim \frac{\beta_{bl2} - \beta_{bl1}}{l_m}$. The gradient is higher in candidates with

bigger $\bar{D}_0$. It is not clear now if the constant value $\beta_{bl2} = 51^\circ$ is universal or is result of coincidence.

Table 3

Stage's candidates with different relative inlet diameters

$\bar{D}_0$	$A_d = \bar{D}_0 / \bar{D}_{0wmin}$	$\beta_{bl1}, ^\circ$	$\beta_{bl2}, ^\circ$	w'_{1s} / u_2	ζ_{fr}	ζ_{mix}	$\eta, \%$
0.53	0.845	40.5	51	0.744	0.0553	0.0181	85.9
0.54	0.881	38.8	51	0.736	0.0578	0.0156	86
0.55	0.897	37.1	51	0.730	0.0607	0.0140	86.1
0.56	0.914	35.4	51	0.724	0.0640	0.0131	86.1
0.57	0.930	33.9	51	0.722	0.0673	0.0118	86
0.58	0.946	32.4	51	0.719	0.0713	0.0117	85.8
0.59	0.963	31	51	0.687	0.0756	0.0119	85.6
0.60	0.979	29.7	51	0.719	0.0801	0.0127	85.3
0.61	0.995	28.4	51	0.719	0.0857	0.0131	85
0.62	1.011	27.3	51	0.722	0.0913	0.0137	84.8
0.63	1.028	26.1	51	0.725	0.0977	0.0151	84.2

The optimal position of the blade leading edge $\bar{l}_{mbl}$ was determined by comparison of candidates presented in the Table 4.

Table 4

Stage's candidates with different position of a leading edge

$\bar{l}_{mbl}$	$\beta_{bl1}, ^\circ$	$\beta_{bl2}, ^\circ$	w'_1 / u_2	ζ_{fr}	ζ_{mix}	$\eta, \%$
0.61	30.7	59.5	0.730	0.0522	0.0110	86
0.64	31.4	57.3	0.726	0.0542	0.0103	86.1
0.67	32.2	55.9	0.724	0.0560	0.0109	86.1
0.7	33	54.3	0.723	0.0579	0.0105	86.2
0.73	33.8	53	0.721	0.0600	0.0111	86.2
0.75	34.3	52	0.722	0.0615	0.0109	86.2
0.76	34.6	52	0.723	0.0619	0.0122	86.1
0.792	35.4	51	0.724	0.0640	0.0130	86.1
0.81	35.9	50.5	0.726	0.0651	0.0131	86
0.84	36.8	49.5	0.730	0.0673	0.0135	85.9

The optimal combination of $w'_{1s} / u_2, \zeta_{mix}, \zeta_{fr}$ corresponds to the range of $\bar{l}_{mbl} = 0.7 - 0.75$. The blade load is changing sufficiently in the studied range of $\bar{l}_{mbl} = 0.61 - 0.84$. May be comparison of candidates with different blade number for $\bar{l}_{mbl} = 0.61$ and 0.84 could correct the presented results to some extent.

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CONCLUSION

The analysis of a typical high flow rate stage of an industrial compressor by CFD, Q3D and engineering-type computer program calculations demonstrates that some design recommendations well-proven on mean flow rate stages with 2D impellers are valid. The principle of 3D blades' design by Q3D non-viscid velocity diagrams optimization is valid. The new solutions and recommendations are proposed in other cases – symmetric sloping tapering of VLD is preferable, better crossover configuration is proposed, coefficient of an optimal inlet diameter A_{dopt} appeared to be about 0.85 but not about of 1,0 as for 2D impellers. The principles of optimization can be applied to stages with other design parameters that will continue the process of design recommendations proof and correction.

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