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CENTRIFUGAL COMPRESSOR STAGE PRELIMINARY AERODYNAMIC DESIGN AND COMPONENT SIZING

Ronald H. Aungier
Product Development
Elliott Company
Jeannette, Pennsylvania

ABSTRACT

Procedures are presented for the preliminary aerodynamic design of centrifugal compressor stages. Methods for selecting the key aerodynamic and geometrical design parameters are based on actual stage design experience, supplemented by performance analysis results. Basic component geometry sizing procedures are presented for impellers, vaneless diffusers, vaned diffusers, crossover bends, return channels and volutes. The preliminary stage designs feature well matched components and practical component geometry. The stage geometry is defined in sufficient detail to permit evaluation by a mean streamline performance analysis and to provide the initial geometry for refinement by detailed aerodynamic design procedures. A preliminary stage design can typically be generated in a matter of minutes. Sample designs are evaluated with a performance analysis to demonstrate the close agreement between design objectives and predicted performance.

NOMENCLATURE

A	passage area, and ellipse axial semi-axis
A_R	vaned diffuser area ratio
a	sound speed
B	ellipse radial semi-axis
B_A	fractional aerodynamic area blockage
B_M	fractional blade metal area blockage
b	passage width (hub-to-shroud)
C	absolute velocity
c_f	skin friction coefficient
d	diameter
H	total enthalpy
i	incidence angle, $(\beta - \alpha)$
K	parameter defined in equation (14)
K_λ	parameter defined in equation (4)
L	impeller axial length, also vaned diffuser blade loading parameter
L_B	blade mean camberline length
m	meridional coordinate
$\dot{m}$	mass flow
Q_0	volume flow $= \dot{m} / \rho_{t0}$
R_c	mean stream surface radius of curvature
r	radius
SP	volute sizing parameter
U	impeller blade speed $= \omega r$
W	relative velocity
Z	number of vanes
z	axial coordinate
α	flow angle with tangential direction
β	blade angle with tangential direction
η	efficiency = stage head/work input head
Θ	polar angle
$2\Theta_C$	vaned diffuser effective divergence angle
λ	impeller tip distortion factor $= 1 / (1 - B_{A2})$
μ	head coefficient, $\eta\,(H_2 - H_0) / U_2^2$
μ_{par}	parasitic work coefficient
μ/η	work input coefficient, $(H_2 - H_0) / U_2^2$
$(\mu/\eta)_B$	impeller blade work input coefficient
ξ	normalized meridional distance
ρ	gas density
σ	slip factor
ϕ	stage flow coefficient $= Q_0 / (\pi r_2^2 U_2)$
ω	rotation speed (radians/sec)

Subscripts

CO	crossover bend parameter
EX	return channel exit turn parameter
H	hub contour parameter

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I impeller parameter
m meridional component
p polytropic condition
S shroud contour parameter
t total thermodynamic condition
u tangential component
VLD vaneless diffuser parameter
VD vaned diffuser parameter
0 stage inlet parameter
1 impeller blade inlet parameter
2 impeller tip parameter
3 vaned diffuser inlet parameter
4 vaned or vaneless diffuser discharge parameter
5 crossover or volute inlet parameter
6 return channel vane inlet or volute discharge parameter
7 return channel vane discharge parameter
8 return channel discharge parameter

Superscripts

* a sonic flow condition
' value relative to the rotating frame of reference

INTRODUCTION

A first step in the aerodynamic design of centrifugal compressor stages is the definition of achievable design goals and the sizing of the stage components. This step provides the candidate stage geometry in sufficient detail to confirm that it can achieve the design objectives via an aerodynamic performance analysis. This can be a very time consuming process, largely based on a trial-and-error procedure. Indeed, the development of efficient detailed aerodynamic design procedures has progressed to a point where the preliminary design can consume a substantial portion of the time required to design a new stage. For a recent design accomplished by this writer, this step accounted for about 25% of the time required to design the stage.

A useful preliminary design system must generate the candidate stage design with minimal input, yet provide sufficient flexibility to meet a wide range of design objectives and constraints. It is a fairly complex process, requiring optimization and matching of several stage components to produce a candidate design for the complete stage. In this writer's experience, attempts to directly design to any arbitrary constraints and performance objectives have been very disappointing. Performance analysis of the preliminary design usually indicates the design objectives have been not met. Often detailed design activity shows that the preliminary design yields geometry that is totally impractical. This really reflects the extreme difficulty in identifying a self-consistent combination of design constraints, performance objectives and key design parameters.

The present paper describes a different strategy for preliminary stage design that avoids most of these problems. Its implementation in a computerized design system has proven quite successful in addressing this aspect of the design process. To be effective, a preliminary design system must be consistent with the detailed design and analysis methods used to evaluate and refine

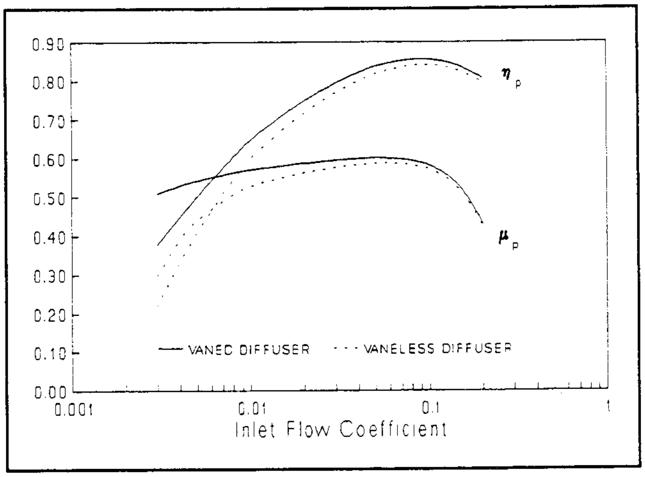

Figure 1: Performance Targets For Covered Impellers

the design. Consequently, this system draws heavily on procedures used by this writer (Aungier, 1988, 1990, 1993a, 1993b). But, the basic approach could be adapted to use other procedures, e.g., those outlined by Whitfield and Bains (1990). The goal is to provide a preliminary stage geometry that can achieve the performance goals, has well matched, optimized stage components and consists of practical component geometry. This system provides the first estimate of each component's geometry for evaluation by aerodynamic performance analysis and for direct refinement with detailed aerodynamic design methods.

THE PRELIMINARY DESIGN STRATEGY

The design system works from reference design conditions that represent good design practice and achievable performance objectives. The designer then modifies this reference design to meet his specific design constraints. To support that process, the present system generates an input file for an aerodynamic performance analysis (Aungier, 1993b) to provide immediate evaluation of the design and direct guidance in adjusting the key design parameters. Experience has shown that iteration between the design system and the performance analysis quickly guides the designer to a consistent set of aerodynamic performance goals and key design parameters for the specific design constraints.

The reference designs are defined by a series of geometric and aerodynamic performance correlations reflecting good state-of-the-art design practice. These correlations were developed from a series of rather successful industrial centrifugal compressor stage designs. These designs include stage flow coefficients from 0.009 to 0.125 and stage pressure ratios up to 3.5. Performance prediction methods (Aungier, 1993b) were used to extend the correlations to cover flow coefficients from 0.003 to 0.2. These reference designs do not attempt to define the best achievable efficiency levels. Rather, they reflect a combination of good efficiency and stable operating range that are considered to be readily achievable with conventional aerodynamic design technology. Figures 1 and 2 show the correlations used to set the reference design performance levels (head coefficient and

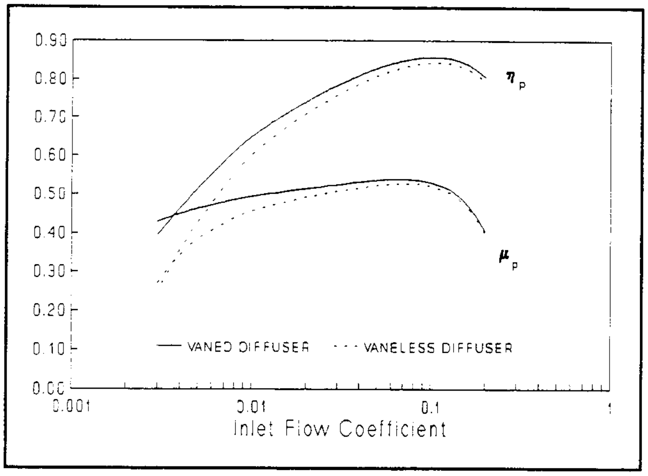

Figure 2: Performance Targets For Open Impellers

efficiency) for covered and open impellers, respectively. Slight differences in the two impeller types are observed, due to differences in the parasitic losses, particularly leakage flow behavior (i.e., eye seal vs. clearance gap leakage). Both vaned and vaneless diffuser stage performance levels are illustrated. These correlations are believed to be conservative for coefficients above about 0.13, where mixed flow designs probably can exceed the levels shown. Pending specific demonstration of the expected higher efficiency levels for mixed flow stages, the more conservative values currently are being used. The correlations used for covered and open impellers are given in equation sets (1) and (2), respectively. The work input coefficient, μ_p/η_p, is the same for either diffuser type in these equations.

$$\begin{aligned} \mu_p / \eta_p &= 0.62 - (\phi / 0.4)^3 + 0.0014 / \phi \\ (\mu_p)_{VD} &= 0.51 + \phi - 7.6\phi^2 - 0.00025 / \phi \\ (\eta_p)_{VLD} &= (\eta_p)_{VD} - 0.017 / (0.04 + 5\phi + \eta_{VD}^3) \end{aligned} \tag{1}$$

$$\begin{aligned} \mu_p / \eta_p &= 0.68 - (\phi / 0.37)^3 + 0.002 / \phi \\ (\mu_p)_{VD} &= 0.59 + 0.7\phi - 7.5\phi^2 - 0.00025 / \phi \\ (\eta_p)_{VLD} &= (\eta_p)_{VD} - 0.017 / (0.04 + 5\phi + \eta_{VD}^3) \end{aligned} \tag{2}$$

The "successful" stage designs described above were used to establish a correlation of impeller axial length, adequate to achieve the above reference design performance levels. To minimize the effect of inducer style, the length is measured from the impeller "eye" or axial entrance.

$$\Delta z_I / d_2 = 0.08 + 1.58\phi \tag{3}$$

Impeller sizing calculations require some key impeller aerodynamic performance data. These include the impeller tip distortion factor, the parasitic losses (windage, disk friction, leakage, etc.) and the eye seal leakage mass flow for covered impellers. Correlations to define these parameters were developed from performance analysis of the "successful" designs. For covered impellers, they are given by

$$K_\lambda = 1 + [0.3 + (b_2 / L_B)^2] \frac{b_2 A_2^2 \sin^2\beta_2}{L_B A_1^2 \sin^2\beta_1} \tag{4}$$

$$\lambda = K_\lambda + (0.00175 / \phi)^2 + 0.0015 / \phi - 0.022 \ln\phi \tag{5}$$

$$\mu_{par} = 0.0014 / \phi \tag{6}$$

$$\dot{m}_{LEK} / \dot{m} = 0.005 + 0.475 / (1 + 500\phi) \tag{7}$$

where equation (4) is a correlation using a form derived from the distortion factor model of Aungier (1993b). For open impellers, λ is increased by 2% over the value predicted by equation (5) to account for clearance effects and

$$\mu_{par} = 0.002 / \phi \tag{8}$$

The impeller inlet relative flow angle, α_1', is set to 30°, which (approximately) minimizes W_1. For vaneless diffuser stages, the other key component flow angles are estimated from

$$\begin{aligned} \tan\alpha_2 &= 0.26 + 3\phi \\ \alpha_4 &= 30^\circ + (\phi / 0.06)^2 \end{aligned} \tag{9}$$

Lower flow angles are used for low flow coefficients to yield wider passages for these high friction loss stages. Similarly, the higher flow angles used for high flow coefficient stages yield narrower passages for stages with high diffusion and hub-to-shroud loading losses. Similar reasoning leads to return channel inlet angles estimated by

$$\tan\alpha_5 = 0.32 + 1.7\phi \tag{10}$$

For vaned diffuser stages, low values of α_2 and α_3 are highly beneficial to good vaned diffuser performance and stable operating range. But, for high flow coefficients, low values of α_2 lead to poor impeller performance. For vaned diffuser stages, the design system uses

$$\begin{aligned} \alpha_2 &= 18^\circ + 0.5\ln\phi + 200\phi^2 \\ \alpha_3 &= 18^\circ + (\alpha_2 - 18^\circ) / 4 \\ \alpha_3 &\leq 18^\circ \end{aligned} \tag{11}$$

and equation (10). It will be seen that vaned diffuser sizing requires $b_3 \leq b_2$ which will override the above value of α_3 for higher flow coefficients. This approach uses the lowest practical value of α_3 consistent with α_2. The blade incidence angles used are $i_1 = 0^O$, $i_3 = -0.5^O$, $i_5 = 4^O$. For impellers and vaned diffusers, a constraint is imposed on the incidence to insure reasonable flow range to choke, i.e.,

$$\sin\beta_1 \geq 1.2\ W_1 /\ a_1^{*'} \qquad \sin\beta_3 \geq 1.2\ C_3\ /\ a_3^* \tag{12}$$

Blade thicknesses, seal geometry, surface finishes and clearances are all set internally, scaled by the impeller diameter. In the interest of brevity, these will not be reviewed, since they will differ for each organization's normal design practice.

The preliminary design system uses these correlated parameters as default values that yield the reference design. They provide a self-consistent set of design specifications for a preliminary centrifugal compressor stage design, as will be shown later in this paper. But, they also represent good design practice, which are usually applicable (or at least good first estimates) to design to other design objectives and constraints.

Other basic design specifications required to size a centrifugal compressor stage must be supplied by the designer, including

- Stage inlet thermodynamic conditions
- Gas equation of state data
- Rotation speed
- Impeller tip diameter
- Stage mass flow
- Definition of component types
- Minimum (shaft) diameter (optional)
- Maximum (casing) diameter (optional)

When design constraints permit, this yields the candidate stage design directly, typically in a couple of minutes. Indeed, this design system has proven to be a useful tool for a quick evaluation of achievable performance levels and the basic stage configurations appropriate for proposed new applications. As required, the designer has the option to modify any of the default data to customize the design to meet specific design objectives or constraints.

IMPELLER SIZING

The design system treats two impeller blade styles. Three-dimensional, blades are constructed with straight-line surface elements connecting the general hub and shroud surface blade designs. This blade style is used for either full-inducer or semi-inducer impellers. Impellers with no inducer use two-dimensional, axial-element blades. Full-inducer impellers assume an axial inlet with uniform inlet flow, while semi-inducer and no-inducer styles correct the leading edge meridional velocity distribution for local passage curvature with a simple potential flow assumption

$$C_{mH}\Delta m_H = \bar{C}_m \Delta\bar{m} = C_{mS}\Delta m_S \tag{13}$$

where Δm is the local contour lengths between successive passage

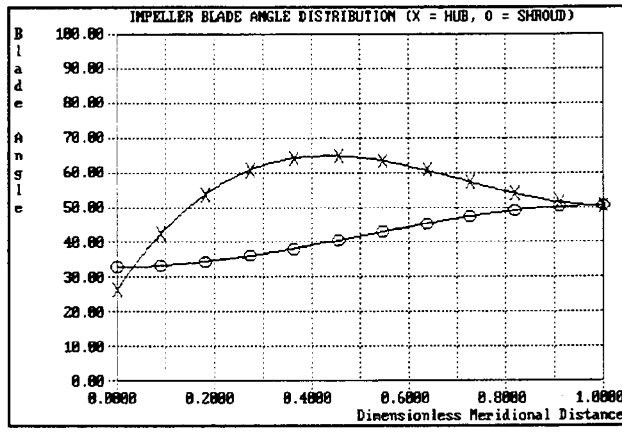

Figure 3: Typical Blade Angle Distributions

quasi-normals for the hub, mean and shroud contours. The impeller design includes an approximate blade design to ensure that practical blade rake angles (i.e., the angles between the blade leading and trailing edges and the meridional plane) can be achieved with the candidate design. Unacceptable rake angles is probably the most common inconsistency encountered in the detailed aerodynamic design of impellers from a preliminary design, particularly for semi-inducer stages. When it is encountered, it can completely invalidate the preliminary stage design. Two-dimensional, axial-element blades are constructed with circular-arc mean camberlines. For three-dimensional blade styles, the generalized hub and shroud blade angle distributions shown in equation set (14) are used for this purpose.

$$\begin{aligned}
\beta_S &= \beta_{1S} + (\beta_2 - \beta_{1S})(3\xi^2 - 2\xi^3) \\
\beta_H &= \beta_{1H} + A\xi + B\xi^2 + C\xi^3 \\
\bar{\beta}_H &= 90K + (1-K)(\beta_2 + \beta_{1H}) / 2 \\
A &= -4(\beta_2 - 2\bar{\beta}_H + \beta_{1H}) \\
B &= 11\beta_2 - 16\bar{\beta}_H + 5\beta_{1H} \\
C &= -6\beta_2 + 8\bar{\beta}_H - 2\beta_{1H}
\end{aligned} \tag{14}$$

The parameter, K, adjusts the hub blade angle at mid-passage to permit control over the blade rake angles. These rake angles are set equal and opposite in sign with a target to hold them less than 15^O but with K limited to maximum of 1. The leading edge blade angles are set from the local relative flow angles assuming a linear variation of blade angle from hub to shroud (a reasonable assumption for modest rake angles). The construction matches i_1 and requires a shroud incidence angle 25% of the hub incidence angle. Figure 3 illustrates typical blade angle distributions generated with these equations.

Hub contours are constructed with the largest circular-arc permitted (with linear extensions where needed). This contour is completely defined by the impeller eye and tip slopes and coordinates. Shroud contours are constructed with three-point cubic spline fit curves. This contour is completely defined by its end point coordinates and slopes and the coordinates of one intermediate point. For cases where the hub contour requires a linear extension at the tip, experience has shown that a corresponding linear extension should be used on the shroud contour to obtain reasonable passage area distributions. This construction permits direct control over the passage area at three locations in the passage. Areas are always set at the blade leading and trailing edges and at the impeller eye. In the case of full-inducer stages, (where two of these locations are identical), the third location where the passage area is set is at mid-passage.

The impeller design is an iterative process where the blades, hub and shroud contours are designed to meet the following constraints

- A linear variation in passage area.
- Match the blade leading edge incidence and flow angles.
- Match the trailing edge velocity triangle.
- Control the leading and trailing edge rake angles.
- Conserve mass at the leading and trailing edges.
- Constrain the average passage width-to-radius of curvature ratio (b/R_c) to a maximum of 1.

The last constraint will yield a mixed-flow design for very high flow coefficient stages. For no-inducer or semi-inducer stages, additional constraints are imposed to locate the blade leading edge. They are

- Limit the magnitude of hub and shroud incidence angles.
- Impose a 5% reduction in passage area between the leading edge and eye positions.

For semi-inducer and no-inducer impellers, the impeller sizing calculations seek to locate the leading edge as far upstream as possible, consistent with the incidence and rake angle limits.

The impeller blade work input and tip tangential velocity are given by the stage work input and parasitic losses.

$$(\mu / \eta)_{blade} = \mu / \eta - \mu_{par} = C_{U2} / U_2 \qquad (15)$$

The tip blade angle is computed by iterative solution of equations (16) and (17), using the specified value of α_2.

$$\sigma = 1 - \sqrt{\sin\beta_2}\sin\alpha_c / Z^{0.7} \qquad (16)$$

$$(\mu / \eta)_{blade} = \sigma(1 - \lambda C_{m2}\cot\beta_2 / U_2) \qquad (17)$$

where the slip factor model used is by Wiesner (1967). Similarly, the leading edge mean meridional velocity and blade angle are computed directly from the specified values of α_1', i_1, and the local blade speed.

$$C_{m1} = U_1\tan\alpha_1 \qquad (18)$$

The inlet passage width is then computed from conservation of mass. The number of required blades is estimated from the blade loading parameter.

$$\Delta W / \bar{W} = \frac{2\pi d_2 C_{u2}}{Z(W_1 + W_2)} \qquad (19)$$

$$\Delta W / \bar{W} \leq 0.9$$

The impeller iteration procedure continues until both the leading edge sizing and the number of blades have converged.

VANELESS DIFFUSER SIZING

Vaneless diffuser sizing is accomplished following the design procedure of Aungier (1993a). The discharge radius is computed to match the maximum (casing) radius, if specified, with allowance for the crossover or volute to follow. Alternatively,

$$r_4 = (1.55 + \phi)r_2 \qquad (20)$$

The vaneless diffuser discharge width is sized to yield specified α_4. A correction to tangential velocity for wall shear effects is computed from a simple form of the conservation of angular momentum equation.

$$\ln[(r_4 C_{u4}) / (r_2 C_{u2})] = \frac{-c_f(r_4 - r_2)}{\bar{b}\sin\bar{\alpha}} \qquad (21)$$

Then, conservation of mass yields b_4. The analysis imposes the requirement that $b_4 \leq b_2$. Any adjustment in passage width between the impeller tip and diffuser exit is imposed on the shroud wall. For highly mixed flow stages, the slope of the hub contour at the impeller tip will be non-radial. A gradual turn of the hub contour to radial is imposed. The non-radial portions of the hub and shroud contours are imposed by circular-arc segments, extending over the same portion of the passage length. This uses a minimum radius of curvature equal to b_2, if possible, but is limited to the first 50% of the passage length. The minimum radius of curvature will always occur on the shroud. The hub radius of curvature will always be larger and is simply set to match any adjustment required.

VANED DIFFUSER SIZING

Vaned diffuser sizing is accomplished following the design procedure of Aungier (1988). The maximum diffuser exit radius is computed in the same manner as used to set the vaneless diffuser exit radius. The vane leading edge radius is estimated by

$$r_3 / r_2 = 1 + \alpha_3 / 360 + M_2^2 / 15 \qquad (22)$$

The Mach number dependence allows additional vaneless space to diffuse high Mach number flows. The passage width, b_3, is computed from conservation of mass to match the specified α_3, where C_{u3} is calculated analogous to equation (21), with the constraint that $b_3 \leq b_2$. The specified vane incidence angle then yields β_3. The number of vanes is selected from resonance considerations as well as aerodynamics. The preferred choice is

$Z_{VD} = Z_I \pm 1$. A low value of Z_{VD} is very beneficial to the stall incidence range, so $10 \leq Z_{VD} \leq 20$ is required. If the preferred numbers are not in this range, $|Z_{VD} - Z_I| \geq 8$ is required, leaving final selection of Z_{VD} and Z_I to the detailed design phase. Discharge sizing is based on the equivalent divergence angle, blade loading parameter and area ratio, with $b_4 = b_3$.

$$\tan\theta_c = \pi(r_4\sin\beta_4 - r_3\sin\beta_3) / L_B \tag{23}$$

$$L = \frac{2\pi(r_3C_{u3} - r_4C_{u4})}{ZL_B(C_3 - C_4)} \tag{24}$$

$$A_R = r_4\sin\beta_4 / (r_3\sin\beta_3) \tag{25}$$

Appropriate design limits (Aungier, 1988) are $L \leq ⅓$, $2\Theta_C \leq 11°$. The process involves scanning candidate values of $2\Theta_C$ from $10.5°$ to $7°$. For each Θ_C, values of A_R from 2.4 to 1.4 are scanned, subject to the constraint that r_4 can not exceed the value estimated above. The first combination to yield $L \leq ⅓$ is the selected design. On occasion, no suitable choice will be found, requiring the designer to modify the impeller tip velocity triangle. When the maximum (casing) radius is specified, this process may yield a value of r_4 less than the value required to match it. In that case, a vaneless passage follows the vaned diffuser match that constraint. The diffuser vane is designed with the mean camberline of Aungier (1988). The exit flow angle, α_4, is computed using the deviation angle model of Aungier (1990). Figure 4 shows a typical vaned diffuser design.

RETURN SYSTEM SIZING

The design of the crossover bend and return channel follows the design procedure of Aungier (1993a). The crossover bend discharge passage width is set by

$$b_6 = b_4\tan\alpha_4 / \tan\alpha_6 / B_{M6} \qquad b_4 \leq b_6 \leq 2b_4 \tag{26}$$

which may override the specified α_6 due the limits imposed. The radius of curvature for a circular-arc hub contour is given by

$$R_{cH} = (b_6 + b_4) / 2 \qquad R_{cH} \geq 0.8(b_8 - b_6) \tag{27}$$

where b_8 is the passage width for the eye of the next impeller, computed from the stage performance targets and mass flow. This insures adequate axial length to accommodate the return system and reasonable values of b/R_c through the crossover bend. The crossover shroud wall is an elliptical contour with axial and radial semi-axes given by

$$A_{CO} = R_{cH} + (b_4 + b_6) / 2 \qquad B_{CO} = R_{cH} + b_4 \tag{28}$$

If the stage maximum (casing) radius is specified, the design system will have constrained the diffuser exit radius to insure this crossover will match that value. The exit turn uses circular-arc hub and shroud contours defined by

$$R_{cEX_S} = b_8 \qquad R_{cEX_H} = 2b_8 \tag{29}$$

The return channel vane employs the mean camberline and thickness distribution of Aungier (1993a). The vane inlet angle, β_6, is set to match the assigned values of α_6 and i_6 and $\beta_7 = 90°$. Since the blade metal blockage, B_{M6}, in equation (26) depends on β_6, a simple iteration process is used to converge on that parameter.

The return channel passage is constructed with a vertical shroud wall. This return system sizing procedure is fully consistent with the detailed design procedure (Aungier, 1993a), but uses a simplified form of construction relative to that procedure. The preliminary design system can set up a data file for the detailed design procedure for refinement of the design using the more general construction which it offers. A typical cross-section of a return channel stage generated with the preliminary design

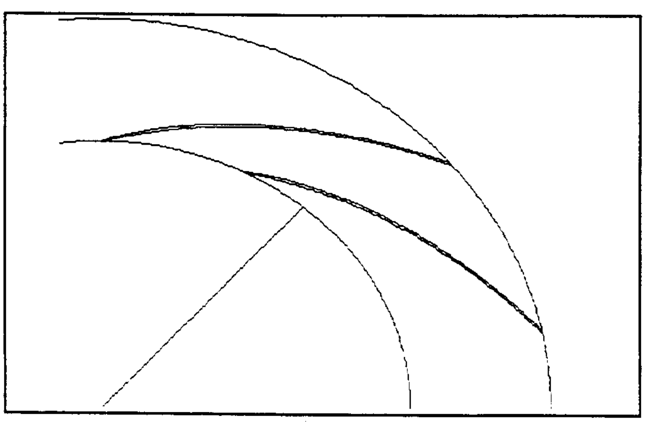

Figure 4: A Typical Vane Diffuser Design

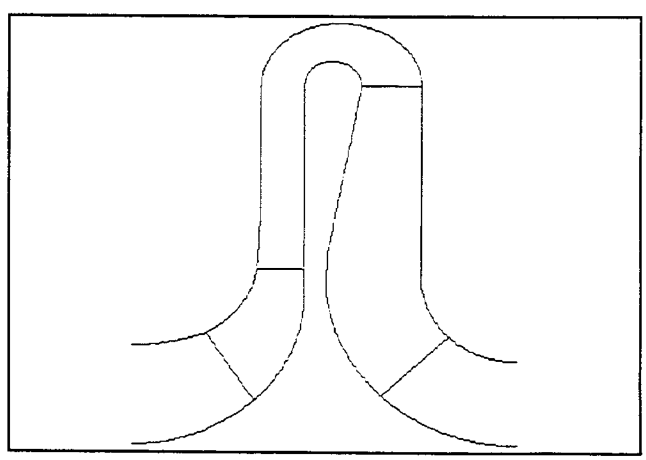

Figure 5: A Typical Return Channel Stage Design

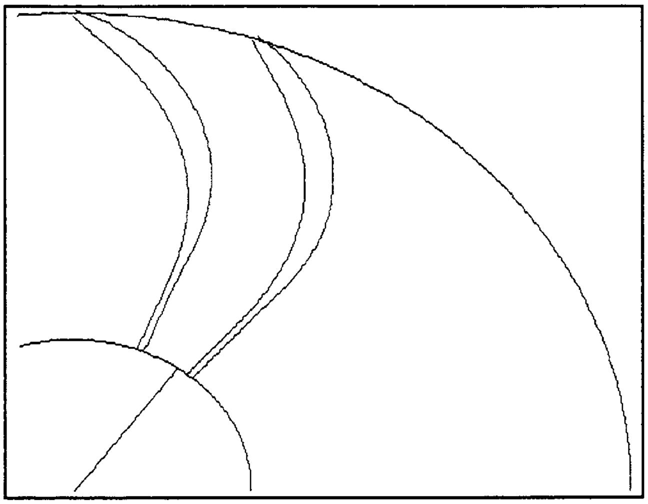

Figure 6: The Return Channel Vanes

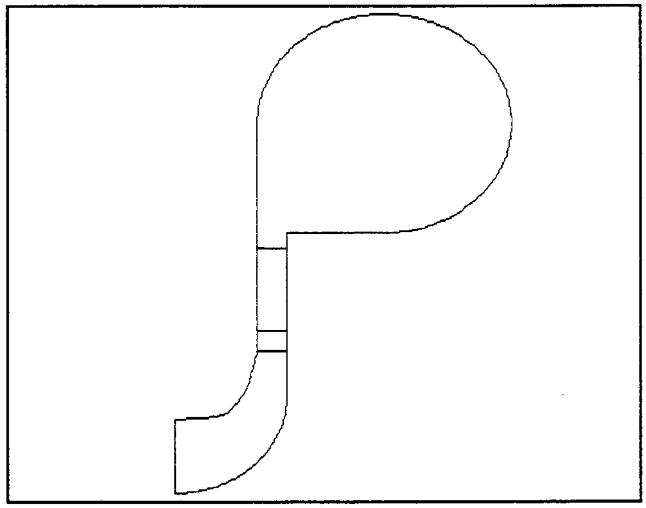

Figure 7: A Typical Volute Stage Design

system is shown in figure 5. Figure 6 shows the return channel vane design for this case.

VOLUTE SIZING

The preliminary design system designs volutes of either elliptical or rectangular cross section. The cross-section shape is significant only when the user has specified a maximum radius for the stage. The volute area is scaled by the volute sizing parameter. The default value is 1.0, but the user may specify an alternate value. For SP = 1.0, the volute cross-sectional area is sized to conserve angular momentum from the diffuser exit. The area and mean radius at the circumferential position where all of the flow has been collected, A_6 and r_6, are the critical parameters. When no maximum radius is imposed, a square or circular cross-section located above the diffuser discharge is used. With a maximum radius defined, the rectangular or elliptical cross-sections are used, with $r_6 = r_4$, with an axial-to-radial aspect ratio up to 1.5 permitted. During the diffuser sizing process, the diffuser exit radius is constrained to insure the volute will fit within the maximum radius. The full-collection area is given by

$$A_6 = SP(A_4 r_6 \tan\alpha_4 \ / \ r_4) \tag{30}$$

and r_6 is a function of r_4, A_6 and the aspect ratio. An iterative process is used to converge on these two parameters. Figure 7 shows a typical volute stage designed by this preliminary design procedure, where a vaned diffuser was used.

IMPLEMENTATION OF THE DESIGN SYSTEM

This preliminary design procedure is implemented as an iterative computer program for personal computers. Extensive use of monitor screen graphical output assists the designer in evaluating the design (e.g., figures 3 through 7 are direct copies of screen graphics from the program). The performance analysis program to which it is interfaced is also run on personal computers. This allows the designer to quickly alternate between these programs to evaluate or customize the candidate centrifugal compressor stage designs. The design program also includes the capability to create input data files for detailed aerodynamic design systems for vaneless diffusers and return channels (Aungier, 1993a), vaned diffusers (Aungier, 1988), and impellers, all of which also run on personal computers. When a suitable preliminary design is established, the geometry can be supplied directly to the detailed aerodynamic design procedures for refinement by those more general design systems.

To demonstrate this design system, a series of eighteen stages were designed over a range of stage flow coefficients and analyzed with the performance analysis. These stage designs employed no-inducer impellers for $\phi \leq 0.05$ and semi-inducer impellers for all others. All designs used a return channel discharge. The design system's default design parameters were used in all cases (i.e., the design system's reference designs were used). The complete design and analysis of one of these designs is easily accomplished in under two minutes. Since half of these designs involved only changing the diffuser type, this entire set of eighteen stage designs and performance analyses was processed in less than fifteen minutes. Figures 8 and 9 compare the performance prediction results with the performance targets as supplied in figures 1 and 2. Note that the preliminary design system produced designs having excellent potential to achieve the target performance levels in almost all cases. In all cases, the performance analysis indicated that the components were well matched at the stage's design flow coefficient. Off-design performance predictions showed that a good stable operating flow range should be achieved by all designs.

Figure 9 shows a local weakness in the design procedure for ultra-low flow coefficient vaned diffuser stages. This is really a weakness in the vaned diffuser design procedure (Aungier, 1988). For these stages, wall shear forces in a vaneless diffuser yield nearly as much reduction in angular momentum as the candidate vaned diffusers. This indicates that aspect ratio effects (not modeled in the design procedure) are quite significant. To design

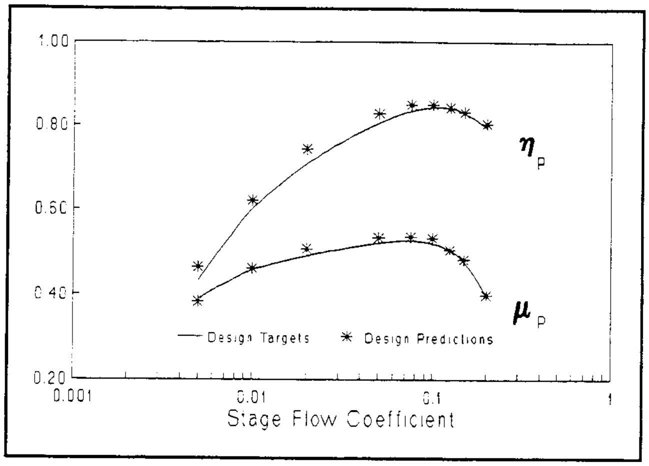

Figure 8: Typical Vaneless Diffuser Stage Results

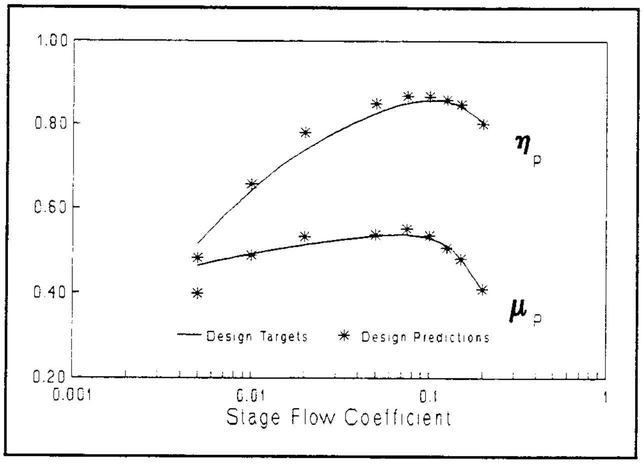

Figure 9: Typical Vaned Diffuser Stage Results

effective vaned diffusers for ultra-low flow coefficient stages, some modification of the vaned diffuser design parameters (L and $2\theta_c$) or their design limits is needed. This would probably increase the vane diffusion limits employed to account for the portion of the diffusion process supported by wall shear. A correction along the lines of equation (21) could probably be used for this purpose. Clearly, this vaned diffuser procedure should not be used for stage flow coefficients less than about 0.01.

The results also indicate no merit in using vaned diffusers for ultra-high flow coefficient stages. This is a direct consequence of the higher impeller tip flow angles used for these stages, leaving very little opportunity for an effective vaned diffuser design. Attempts to design to lower impeller tip flow angles produced lower predicted impeller performance, suggesting that vaned diffusers are, indeed, of little merit in these cases.

CONCLUSIONS

A systematic preliminary aerodynamic design procedure for centrifugal compressor stages has been presented. It treats impellers, vaneless diffusers, conventional vaned diffusers, crossover bends, return channels and volutes. This procedure provides candidate stage geometry for evaluation with a performance analysis and for refinement using detailed aerodynamic design procedures. These candidate designs feature well matched stage components and practical geometry.

Based on test results from successful stage designs, recommended values of key component design parameters and consistent stage performance targets are presented for a wide range of stage flow coefficients. These "reference" designs represent performance levels considered to be readily achievable with conventional aerodynamic design technology. Performance analysis of these reference designs confirms their potential to achieve the target performance levels. One exception is ultra-low flow coefficient vaned diffuser stages where the vaned diffuser design procedure of Aungier (1988) appears to require revision.

Direct use of these reference designs can provide a candidate stage design and its performance analysis in less than two minutes. An unexpected benefit is the design system's capability to evaluate design options. It has proven to be a very effective tool for evaluating achievable performance levels and acceptable design constraints for new stage designs or applications.

When design objectives or constraints require, this design system permits the designer to modify any of the key design parameters to modify the reference design. Experience has shown that this process is much more direct and reliable than attempting to "guess" a complete set of stage design parameters for the specific design goals. Simple interfacing between the preliminary design system and an aerodynamic performance analysis, provides direct guidance to systematically refine these alternate design parameter estimates to achieve a good candidate design for the specific design constraints.

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