
SHAPES OF BLADES AND FLOW PARTS OF TURBOMACHINES

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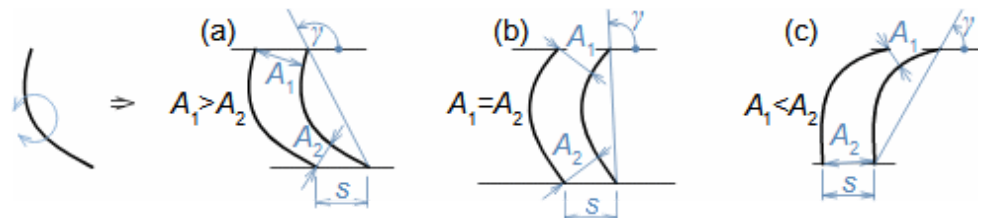
TYPES OF PROFILE CASCADES

The size of the inlet and outlet flow area of the blade cascade is crucial for the resulting working fluid velocities. From this, we distinguish three types of profile cascades depending on whether the blade passages create confuser, diffuser (convergent or divergent) passages, or passages with the same flow area between the inlet and outlet.

Changing type of profile cascade by changing stagger angle of profile

The shape, respectively the inlet and outlet flow area of the profile cascade, is determined by the stagger angle (inclination in the cascade). Any basic type of profile cascades with the same pitch can be constructed from the same profile by simply turning the profile or changing the stagger angle of the profile in the cascade, see FIGURE 312.

312:



(a) confuser cascade - also called overpressure, turbine cascades; (b) pressureless cascade - with same flow area pressure and velocity between inlet and outlet do not change; (c) diffuser cascade - also called overpressure, compressor or turbine cascades in case A_1 is critical flow area. A [m^2] flow area. γ [$^\circ$] stagger angle of profile in cascade; s [m] pitch. The profiles are simplistically drawn as if they were made of sheet.

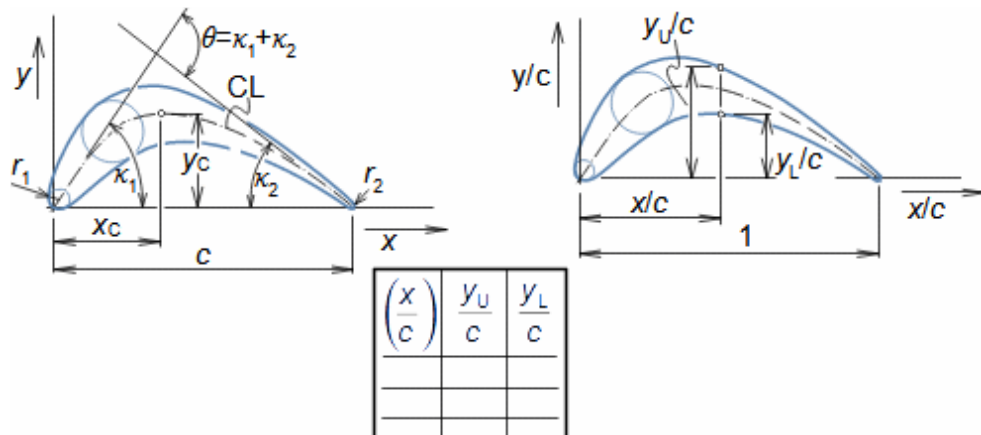
BLADE PROFILE

The profile of the blades is usually shaped like a cambered drop. The camber of the blade is based on the requirements for the camber of the fluid velocity inside the blade passages. However, the shape and size of the profile also depend on the strength and other requirements related to the operation of the turbomachine.

Profile drawing

The blade profile can be recorded using blade profile geometry rules. The choice of the method of recording the blade profile shape depends on the medium of this recording, or the most suitable method in relation to the required form of production documentation. Currently, graphical output (using vector graphics) is sufficient, e.g. in CAD systems, as machine tools and computing software are able to work directly with such output, but other forms of blade profile notation exist. For example, they are written in tabular form in the x ; y coordinates and using the shape of the camber line of the profile, see FIGURE 314 (p. 4).

314:



CL-camber line of profile (geometric location of centres of circles inscribed in profile). $\theta = \kappa_1 + \kappa_2$ [°] camber; y_c [m] maximum camber; x_c [m] position of maximum camber; κ_1, κ_2 [°] angles of camber line (at leading edge of profile and trailing edge of profile); c [m] chord; r [m] radius.

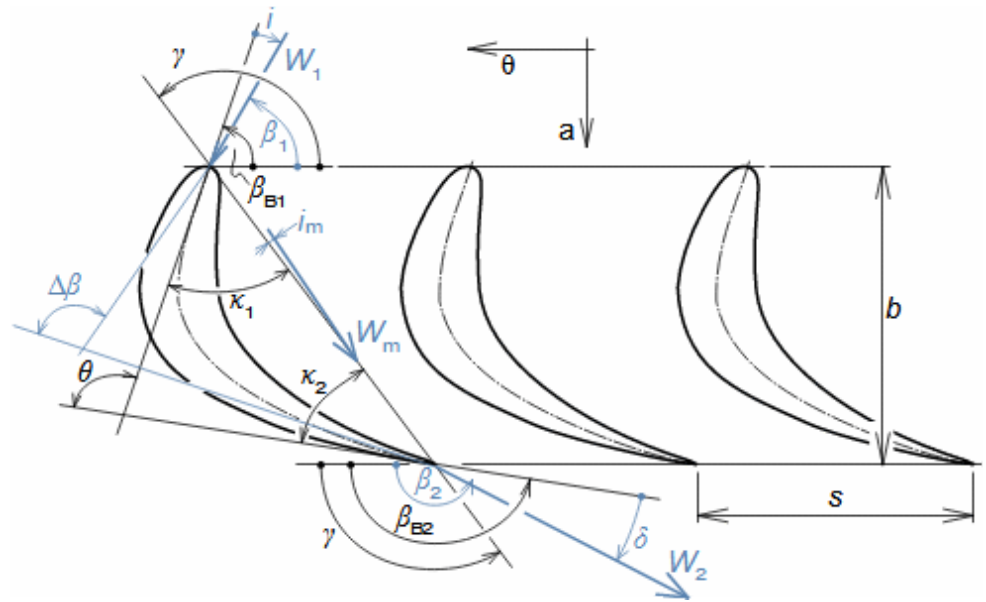
Shape of Camber line

The shape of the camber line of the profile is most often formed by parts of a circle, parabola, logarithmic curve and other types of curves (or two curves with a common tangent at the point of intersection). Typical values of the x_c/c ratio are between 0,4 and 0,5, with pressureless cascades the ratio tends to be around 0,5.

Relationship between camber line and camber of flow in geometric and aerodynamic characteristics of profile cascade

The camber line of the profile, respectively the camber of the profile, should copy the expected streamlines inside the blade passages. These can be approximately determined with the required camber of the relative velocity, which is based on the calculation of velocity triangles. The relationship between geometric and aerodynamic variables is referred to as the geometric and aerodynamic characteristics of the profile cascade and can be graphically represented using FIGURE 315 (p. 5) – from this, mathematical relationships can also be derived, see PROBLEM 755 (p. 5). The camber of a blade and a camber of the flow with a properly designed blade profile has approximately the same value. For this, it is necessary that the required velocity at the inlet to the blade cascade must maintain some angle of attack with the camber line, because at the outlet of the profile cascade the direction of relative velocity deviates from the camber line by a deviation angle, FIGURE 315 (p. 5).

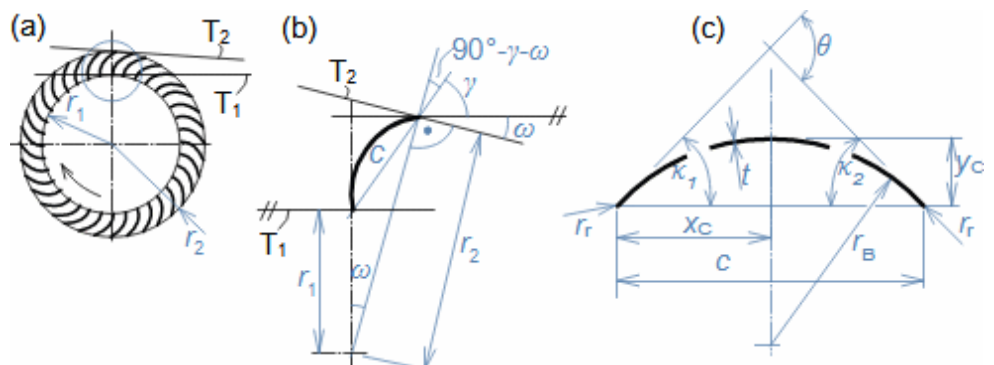
315:



β_{B1}, β_{B2} [°] inlet and outlet angle of blade profile; i [°] angle of attack; δ [°] angle of deviation; $\Delta\beta$ [°] camber of flow; b [m] width of profile cascade; W_1, W_2 [m·s⁻¹] attack and outflow velocity; W_m [m·s⁻¹] mean aerodynamic velocity in cascade; i_m [°] angle of attack of mean aerodynamic velocity.

Problem 755:

Design the blade geometry and stagger angle of a low pressure radial fan with forward curved blades. The rotor dimensions are: $r_1=24,6$ mm, $r_2=28,9$ mm, $\beta_1=158,9^\circ$, $\beta_2=18,8^\circ$. The blade is a single thin sheet. The camber line is formed by a circular arc. The design is made for an angle of attack and an angle of deviation of 3° . The solution of this problem is shown in APPENDIX 755.



(a) rotor; (b) stagger angle detail; (c) manufacturing drawing of blade. ω [°] auxiliary angle.

Flow adhesion to profile as a basic aerodynamic requirement

The profile shape is based on the requirements that the blade must meet in the machine, usually the profile resembles a curved drop. The droplet shape allows the ideal adhesion of the fluid to the blade surface and therefore follows the camber of the blade. The property of a fluid to follow a surface is called the Coanda effect, after the Romanian engineer Henri Coandă (1886-1972), who studied the wrapping of surfaces and bodies. The Coanda effect is clearly visible when water is drawn from a cup that does not have a rim around the neck. Up to a certain angle of inclination of the cup, the water runs down the outer surface of the cup instead of flowing directly to the ground. The pressure of the water stream on the surface of the cup is caused by the viscosity, buoyancy, and lower pressure in the flowing liquid than the ambient air pressure. A similar phenomenon occurs when water flows around a cross pipe, for example in a condenser, etc. The adhesion of the fluid to the wrapping surface has limits, beyond which the flow separation from the wrapping surface, these limits are dealt with in the aerodynamics of profile cascades.

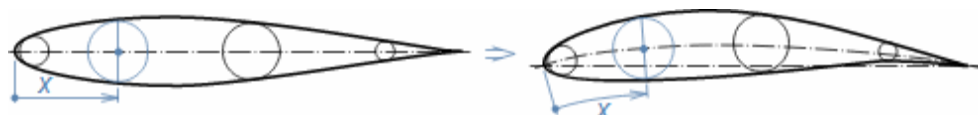
Profile catalogues

A suitable profile can be selected from profile catalogues based on aerodynamic requirements. The shapes and aerodynamic data of thin and low camber profiles can be obtained from extensive airfoil catalogues used in aeronautics, for example [Abbott and Doenhoff, 1959]. If a suitable blade profile is missing from the catalog, it must be developed and experimentally verified.

Developing a new profile using a base airfoil

The development of a new profile is usually based on different camber of the so-called base airfoil, which is a symmetrical smooth airfoil, see FIGURE 618. There are many base airfoils differing from each other in shape, aerodynamic characteristics and other properties, see for example [Abbott and Doenhoff, 1959]. The development of a new profile resulting from the cambering of a base airfoil makes it possible to systematically define the aerodynamic differences between different camber and to clearly catalogue these new profiles according to the base airfoil they are based on.

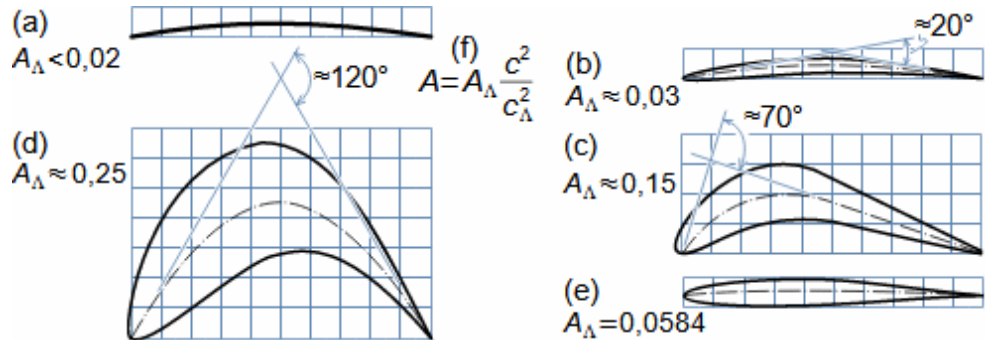
618:



Profile cross-sections at root of blades

In addition to aerodynamic requirements, the airfoil must also meet strength requirements, which affect the required profile thickness at the root of the blades, where the stresses from centrifugal forces and bending are highest. FIGURE 1210 (p. 7) shows typical blade root profiles and their cross-section.

1210:



(a) profile common in radial stages or axial stages with very little camber; (b) thin profile with little camber of flow common in hydraulic machines or turbocompressors (camber line is circle); (c) profile of lightly loaded heat turbine blades (camber line is circle and straight line); (d) profile of heavily loaded heat turbine blade (camber line is parabola); (e) profile of wind turbine blade (NACA 63-209); (f) Cavalieri formula for recalculating profile cross-section for different chord lengths. A [mm²] actual blade profile cross-section; A_{Δ} [mm²] blade profile cross-section at c_{Δ} chord length (numbers for profiles in this figure are for 1 mm chord length, the so-called specific profile cross-section).

Noise level,
resistance to fouling
and cavitation

A frequent requirement for blade profile characteristics is low noise (fans, wind turbines, etc.), which is mainly influenced by the radii of the leading and trailing edges. Wind turbine blade profiles are also subject to requirements for the least possible fouling (dust) on the surface, which is a function of the shape, roughness and material of the surface. In addition, for hydraulic machines, the profiles are more or less sensitive to cavitation, etc.

RANGE OF VALUES OF SOME GEOMETRIC AND AERODYNAMIC VARIABLES OF BLADE CASCADES

The values of geometric and aerodynamic parameters of the blade cascade are usually based on the requirement for maximum internal efficiency of the stage. This requirement may not always be prioritized; blade cascade parameters can be chosen based on the needs performance of machine and cost.

Number of blades
based on blade
cascade density

The required camber of flow is a function of the camber, of pitch, of chord respectively the number of blades, where the ratio between the length of the chord and the pitch of the profile cascade is referred to as a density of profile cascade, see FORMULA 619.

619:

$$\sigma = \frac{c}{s}$$

σ [1] density of profile cascade.

*Estimation of optimal
density of profile
cascade*

With a higher density of the profile cascade, a greater camber of flow can be expected and vice versa. On the other hand, the number of blades and frictional losses in the cascade increase with the density of the profile cascade. Thus, there is an optimum value for the density of the profile cascade. In the case of diffuser profile cascades, the optimum density should be sought to be one at which the c/a_m ratio is around 2,5, where a_m is the mean width of blade passage [Pfleiderer and Petermann, 2005, p. 408], see FORMULA 323. In the case of confuser cascades, this ratio is usually less than 2,5. These ratios are valid for profile cascades composed of thin, low camber profiles.

323:

$$a_m \approx \frac{s}{2} \left(\frac{1}{\sin \beta_1} + \frac{1}{\sin \beta_2} \right)$$

a_m [m] mean width of blade passage. The derivation of the equation is shown in APPENDIX 323.

*Optimal density of
heat turbine blade
cascades*

The density of profile cascade with highly camber heat turbine profiles can be approximated by the Zweifel coefficient. The Zweifel coefficient $C_{L,0}$ is the ratio of the tangential component of the force on the blade from the fluid flow F_θ to the product of the blade area and the dynamic pressure of the relative velocity at the outlet of the blade cascade, see FORMULA 793. The value of this coefficient for the designed profile cascade should be in the range of 0,75...0,85 for modern profiles with high blade material strength up to 1 [Japikse, 1997, p. 6-17].

793:

$$C_{L,0} = \frac{F_\theta}{\frac{1}{2} \rho \cdot W_2^2 \cdot b \cdot l} = 2 \frac{s}{b} (\cot \beta_1 - \cot \beta_2) \sin^2 \beta_2$$

Definition formula for Zweifel coefficient: $C_{L,0}$ [1] Zweifel coefficient; F_θ [N] tangential component of force on blade acting by flow; l [m] blade height; ρ [kg·m⁻³] density of working fluid. The derivation of the equation is shown in APPENDIX 793.

*Width of profile
cascade*

The width of the profile cascade b , defined in FIGURE 315 (p. 5), is based on the required length of the chord, which is a compromise between the optimal aerodynamic design and the required strength of the blades and their roots. If we know the width, it is not a problem to determine the number of blades respectively the pitch from the density of profile cascade σ .

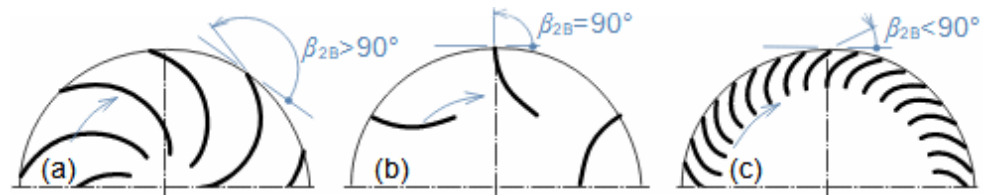
Optimal angles of axial stage for maximum efficiency requirements

When designing the geometric and aerodynamic parameters of axial turbine stages, the goal is to achieve a small output velocity V_2 and the largest possible value of the tangential component of the input velocity V_{10} , respectively the smallest possible profile angle at the stator output, so that the angle of the absolute input velocity α_1 is also as small as possible (for manufacturing reasons, the minimum value of this angle is usually around 8° , depending on the possibility of production and the strength of the blades). The advantage of a smaller angle α_1 is also that for the required velocity component V_{10} , a smaller velocity V_1 is sufficient, thus reducing frictional losses in the stator cascade of blades.

Angles of radial blades are based on requirements for radial stage properties

The properties of the radial stages depend significantly on the profile angle on the rotor circumference (FIGURE 267), see dimensionless characteristics of radial stages. However, the best aerodynamic properties have backward curved blades whose camber line corresponds to a logarithmic spiral, see PROBLEM 274. In the case of turbine radial stages, the angle of the camber line of the profile at the rotor circumference other than 90° is practically only found in Francis turbines.

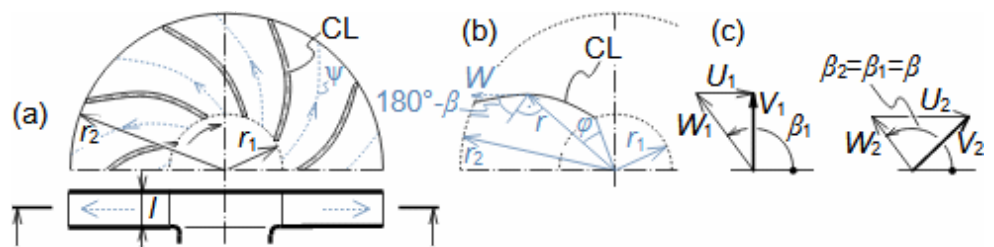
267:



(a) backward curved blades; (b) radial blades; (c) forward curved blades.

Problem 274:

Find the optimum shape of the camber line of the radial fan blade profile with backward curved blades. The given parameters are: $r_1=15,25$ mm, $r_2=30$ mm, $\beta_1=120^\circ$. The solution of this problem is shown in APPENDIX 274.



(a) cross-section of rotor and marking shape of relative velocity streamline; (b) quotation of logarithmic spiral; (c) relation between angles of relative velocity in potential flow through rotor with constant height of blades. ψ -relative velocity streamline. l [m] height of blades; V [$\text{m}\cdot\text{s}^{-1}$] absolute velocity; U [$\text{m}\cdot\text{s}^{-1}$] tangential velocity; W [$\text{m}\cdot\text{s}^{-1}$] relative velocity; β [$^\circ$] angle of relative velocity; ϕ [$^\circ$] angle of logarithmic spiral for its investigated point distant from centre of spiral by r .

*Maximum camber of
flow in diffuser
profile cascades*

Diffuser profile cascades are most commonly used for the working machine stages. Diffuser profile cascades are more sensitive to flow separation from the blade as the camber increases and therefore a small camber of flow $\Delta\beta$ is required, ranging from 15° to 30° . The goal is to achieve the smallest possible V_{10} values for the maximum absolute values of the Euler work.

SHAPES OF BLADES

The shape of the blade determines the final shape of the blade channel. The simplest blade channels have the same shape along the entire length of the blade. Such blade channels can be created using STRAIGHT BLADES – blades with the same shape and profile size along their entire length. In the case of axial stages, the radius and pitch of the profile cascade change. Variable blade channel shapes can be created using TWISTED BLADES, which have a variable profile shape along their length. In addition to these two basic shapes, there are also radial blade shapes for RADIAL STAGES, whereby straight blades are also used for types without axial parts, see FIGURE 267 (p. 9).

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*Straight blades are
based on 1D
calculation of stage*

Straight blades of axial stages are usually used where a small ratio between blade length and mean blade radius can be achieved, so that the spatial character of the flow is not so apparent and 1D calculation is sufficient for stage design. The advantage of straight blades is the simplicity of design, manufacture and cost. They are often manufactured by drawing as circular wires, see FIGURE 278.

278:



left-static steam turbine blade made of drawn profile with machined groove for attachment at the base of the blade; right-rotor steam turbine blade made of drawn profile with machined tip and foot made of forging.

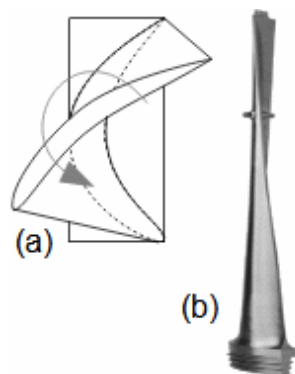
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Twisted blades with
variable stagger
angle along length of
blade

Untwisting of blades,
or changing stagger
angle of profile
during operation

Twisted blades of axial stages are blades with a change of stagger angle of profile in the cascade along the length of the blade and usually with a change of profile. The design of the twisted blade takes into account the spatial character of the flow in the stage and the changes in the velocity triangle and reaction, respectively (see 2D calculation of the reaction of a Kaplan turbine). The resulting blade shape is complex and increases manufacturing costs compared to shaped straight blades (usually produced on a 5-axis milling machine from a shaped casting, but there are other manufacturing technologies for hollow blades).

Strongly twisted and long rotor blades are subject to untwist by centrifugal forces, so-called aeroelasticity, (FIGURE 948a), other deformations are from the working fluid flow. Twisted blades can be strengthened using an integrated blade vibration snubber, which locks into the neighbouring blade at certain machine rotational speeds, thereby preventing further untwisting of the blades, see FIGURE 948b. It is therefore necessary to take into account that the stagger angle of profile geometry will be different at machine standstill than at nominal rotational speed. Untwisting is also evident in wind turbine blades, which also change shape due to axial force from the airflow.

948:

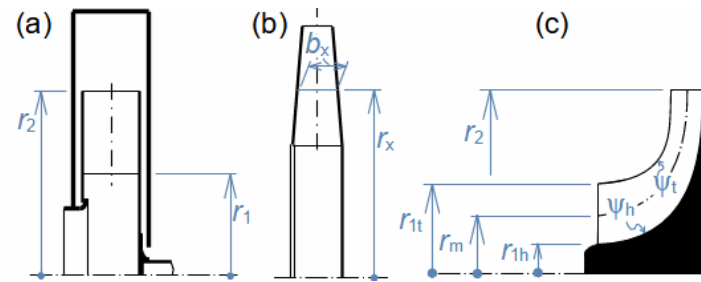


(a) untwisting of twisted blade; (b) blade with integrated vibration snubber

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Shapes of radial
blades

The blades of radial stages are either of such shape that they form purely radial blade passages, or such that they extend into the axial direction (FIGURE 567c, p. 12). In the case of purely radial blades, they are usually straight blades of often constant width, sometimes made of sheet metal (see PROBLEM 755, p. 5). In the case of a constant meridional velocity, the width of the radial blade decreases as the radial coordinate, respective rotor circumference, increases, see FIGURE 567b (p. 12).

567:



(a), (b) examples of r. straight blades; (c) example of r. blades with axial part. The index _t indicates the tip of the blade, _h the root of the blade or the radius of the shaft. b [m] width; r_m [m] mean radius of blade.

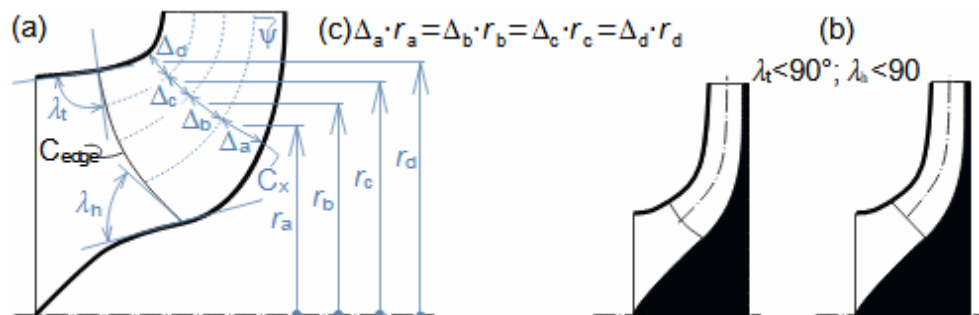
Shapes of meridional
surfaces of radial
stages

Radial blade leading
and trailing edge
shapes

The curves ψ_h and ψ_t (meridional surfaces) at the root and tip of the blades (FIGURE 567c) are either circles or other smooth, easily describable and manufacturable curves.

The axial part of radial blades is used in turbine stages and working machines. In turbines, it has the function of uniformly transferring flow from radial to axial direction with a reduction of tangential velocity component V_{20} . The axial part is called the inducer in the case of working machine stages. In both cases, this part of the radial stage is calculated as twisted (see 2D-calculation of inducer), because there is usually a large difference between the blade speed at the root and tip of the blades. In rotodynamic pumps and Francis turbine, the axial part is usually not pronounced (FIGURE 1015) to avoid large differences in relative velocity angles compared to the optimal angle preventing cavitation. In this case, the blade edges are bounded by the C_{edge} curve. The C_{edge} curve is a smooth curve that follows the radius at the tip r_t approximately perpendicularly ($\lambda_t \approx 90^\circ$), the angle λ_h is usually less than 90° [Pfleiderer and Petermann, 2005, p. 157]. Only for rotors with small relative blade width b is the C_{edge} curve replaced by a straight line (FIGURE 1015b). The drawing of individual streamlines, respectively the boundaries between the individual elementary stages, is based on the simplified rule that for a drawn arbitrary continuous curve C_x (FIGURE 1015a) the EQUATION 1015c holds. The manufacture of this part of the blade by machining is difficult and therefore this type of rotor consists of several parts - see FIGURE 891 (p. 13).

1015:



C_{edge} -curve of leading edge (hydrodynamic pumps) or trailing edge (Francis turbines). The derivation of EQUATION 1015c is shown in APPENDIX 1015.

891:

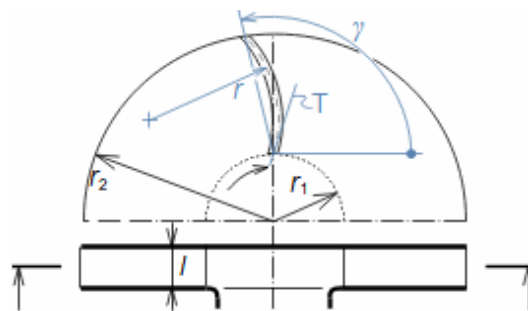

Rotor of radial compressor is glued together from cast inducer (aluminium alloy) and precision forging made of duralumin. Rotor diameter is 160 mm, anodized finish.

*Both blade bladeless
and blade radial
stator cascades are
used*

Radial stages very often contain a bladeless stator. Although the bladeless diffusers have lower efficiency at nominal parameters, they have a smoother efficiency curve when the flow rate changes than a stage with a blade diffuser. Good flow change characteristics can also be achieved with blade diffusers, but at the cost of Turnable blades, which are more technologically demanding and expensive, including the control mechanism. For the same reasons, so-called bladeless diffusers are used for radial turbine stages, or the stator blades are rotatable, as for example in some turbine rotors of turbochargers and Francis turbines. A combination of stator blades and a more higher radial gap between the blades and the rotor is also possible, which acts as the bladeless diffuser (FIGURE 567b, p. 12), or in turbines a bladeless confuser.

Problem 275:

The figure shows a impeller of a respiratory radial fan with backward curved blades printed on a 3D printer. Users of this fan complain about noise. Make suggestions to change the geometry of the blades, which should lead to a reduction in fan noise. The solution of this problem is shown in APPENDIX 275.



T-tangent to the camber line of the profile, which in this case is a circular arc.

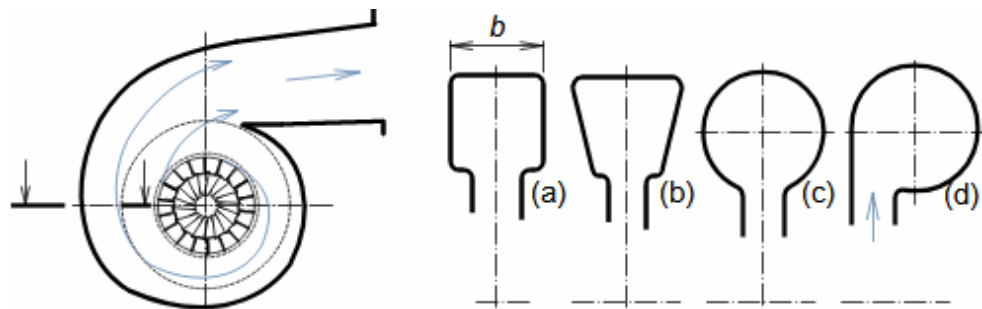
SHAPES OF TURBOMACHINE BRANCHES

The shape of the branch is based on the purpose, the type of machine and especially the direction of the working fluid flow from or to the blade section. If the blade section connects to the branch radially or diagonally, then the branch is of **SPIRAL** design. If the blade section connects to the branch axially, then **AXIAL BRANCHES** are used. In both cases, the shape and size of the branch should be such that the circumference of the connecting blade cascade is at the same pressure. There are no significant density changes in the branches because they are fluid transport passages.

~
*Calculation and
shapes of spiral
casings*

The basic design of the spiral casing is based on the theory of potential flow, in which the streamlines have the shape of logarithmic spiral. The basic spiral casings shapes are shown in FIGURE 754, with some shapes not satisfying the conditions of the potential flow equations, resulting in vortices. However, they have other advantages - in particular, they reduce the calculated diameter of the spiral casing, which results in a much larger diameter than the rotor diameter for a constant casing width and potential flow. Spiral casings can also be reduced by ending them at less than 360° , see PROBLEM 264 - shortened casings can be used where size is a more important parameter than efficiency.

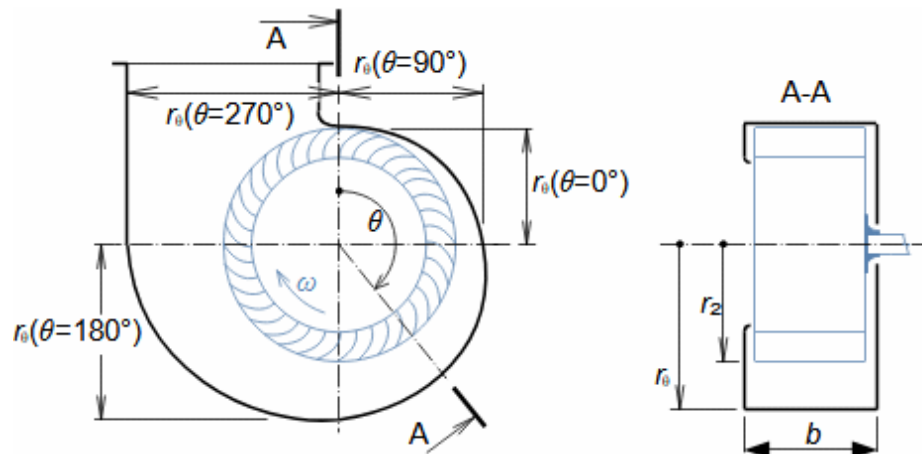
754:



(a) rectangular (constant casing width - mainly used in fans); (b) trapezoidal (gradual extension leads to lower losses than step extension and its shape is very close to the condition for potential flow); (c) circular; (d) tangential outlet casing. b [m] casing width.

Problem 264:

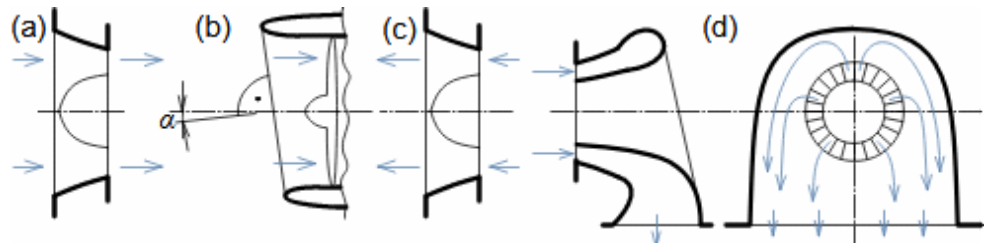
The figure shows a drawing of a spiral casing of a radial fan with forward curved blades, propose the dimensions of this spiral casing. The casing has a rectangular cross-section. The outer radius of the impeller is 28,9 mm, the width of the casing is 23,1 mm, the tangential component of the absolute velocity at the outlet of the impeller is $20,9 \text{ m}\cdot\text{s}^{-1}$ and the airflow is $100 \text{ m}^3\cdot\text{h}^{-1}$. The calculation is carried out for the potential flow case. The solution of this problem is shown in APPENDIX 264.



~
Basic branches
shapes for axial
stages

FIGURE 385 shows some designs of axial and side branches. In the case of FIGURE 385b, the jet engine axial beveled branch that allows for more optimal pressure distribution of the first stage of compressor. The maximum engine power (air consumption) is during takeoff, and therefore the angle α approximately corresponds to the pitch angle during takeoff. More details on this issue, including the calculation of the optimal deflection angle α , are given in [Mattingly et al., 2002, p. 424].

385:



(a) axial inlet (e.g. combustion turbine inlet); (b) jet engine inlet; (c) axial outlet (e.g. axial fan outlet); (d) side branches (e.g. axial compressor side branches). α [°] deflection angle.

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