PRINCIPLES OF OPERATION AND MAIN PARAMETERS
FOR
MERIDIONALLY ACCELERATED FANS
(Centrifugal and mixed flow fans)

1.0 THEORETICAL MODEL FOR ALL FAN TYPES

All fan design is based on Euler's turbine theory and variations thereof, and the various principles, relationships and design techniques involved have been described by Eck[1] and Dixon[2] et al. All of this information has been widely published and circulated amongst practitioners of the technology for more than 90 years. The body of knowledge is continuously expanding through developments in fluid mechanics, thermodynamics and compressible fluid flow.

1.1 Euler's turbine theory

Euler's turbine theory assumes incompressible flow and is applicable to centrifugal, mixed flow and vane axial fans. The general equation used to estimate the theoretical pressure rise generated by any turbomachine contains three terms that involve absolute and relative velocities:

Pressure Rise (Δp_{total}) = $\rho/2 \times \{(c_2^2 - c_1^2) + (u_2^2 - u_1^2) + (w_1^2 - w_1^2)\}$. (See Figure 1 below)

- The term $\rho/2\ (c_2^2 - c_1^2)$ represents the increase in kinetic energy. The conversion of this kinetic energy into static pressure takes place external from the impeller. The term does not include the effect of losses.

- The term $\rho/2\ (u_2^2 - u_1^2)$ represents the increase in static pressure within the impeller due to centrifugal force (meridional acceleration = ω^2/r) and is therefore not affected by losses.

- The term $\rho/2\ (w_1^2 - w_2^2)$ represents the change in static pressure or kinetic energy as the flow passes through the blade passages:

  - If $w_1 > w_2$, this implies that the blade swept area at the inlet is less than the blade swept area at the discharge and the area ratio > 1.00 (typically 1.0 to 1.10 for high specific speed centrifugal fans). In this case the relative flow would be decelerating and this would result in a small increase in static pressure.
  - If $w_2 > w_1$, this implies that the blade swept area at the inlet is greater than the blade swept area at the discharge and the area ratio < 1.00. The relative flow would therefore he accelerating and this is reflected by an increase in kinetic energy. This would be the case with a mixed flow fan, but can also apply to some centrifugal fans.

1.2 Differences between fan types

Centrifugal fans

All centrifugal fan designs makes use of all three terms in Euler's equation. This is illustrated in Figure 1 below. Please note that the vectors are not drawn to scale.

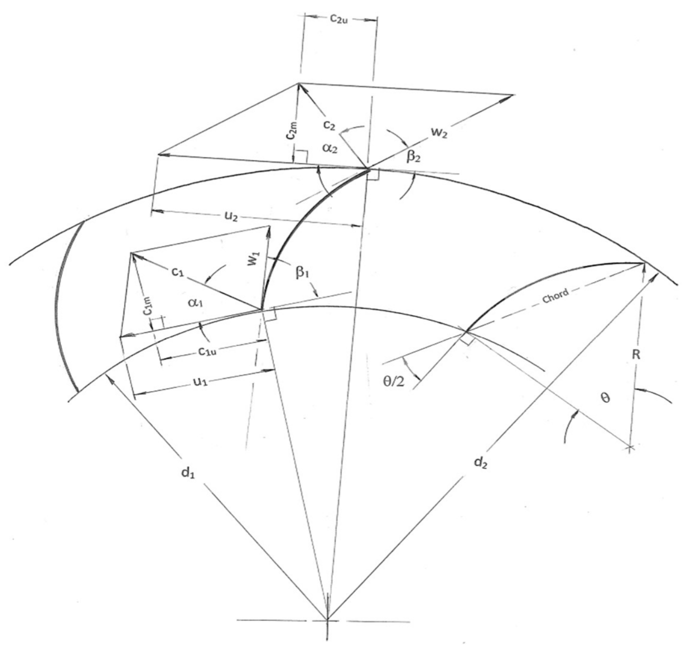


Figure 1 – Velocity diagram for a centrifugal fan

Vane axial fans.

For vane axial fan designs, the diameter of the hub at inlet is the same as at the diameter of the hub at discharge from the impeller ie., $u_2 = u_1$ and $\rho/2\ (u_2^2 - u_1^2) = 0$. This is the reason why axial fans exhibit much lower pressure rise coefficients than centrifugal or mixed flow fans. In this case Euler's equation reduces to:

$\Delta p_{total} = \rho/2 \times \{(c_2^2 - c_1^2) + (w_1^2 - w_1^2)\}$.

Mixed flow fans.

The concept for a high efficiency mixed flow fan design is attributed to Friedrich Schicht who almost 90 years ago was the first to recognise that if a meridionally accelerated axial fan was designed such that $w_1 = w_2$, there would be no change in relative velocity in the impeller and therefore no change in static pressure within the impeller. This eliminates any requirement for the accurately profiled aerofoil blades normally associated with vane axial fans. An additional benefit demonstrated by Schicht was the fact that the output of these axial fans could be effectively varied over the full operational envelope by means of inlet radial vane controls.

This concept was a significant step because the relative velocities inside the impeller passages are high and diffusion losses will accordingly also be high. By eliminating diffusion losses, it became possible for efficiencies of up to 89% - 90% to be achieved.

The main disadvantage of this design is that all static pressure recovery takes place externally, and this demands careful attention to stator vane and diffuser design.

For true Schicht mixed flow fan design, the Euler equation reduces to:

$$\Delta p_{total} = \rho/2 \times \{(c_2^2 - c_1^2) + (u_2^2 - u_1^2)\}.$$

However, it must be noted that most mixed flow fans currently available for mine ventilation incorporate a cylindrical outer casing and this inescapably involves some degree of diffusion through the blade passages ($w_2 < w_1$). For this reason these fans do not meet the requirement for $w_1 = w_2$ and efficiencies are significantly lower (the extent being related to flow diffusion losses in the impeller blade passages).

The pressure generated is therefore governed by:

$$\Delta p_{total} = \rho/2 \times \{(c_2^2 - c_1^2) + (u_2^2 - u_1^2) + (w_1^2 - w_1^2)\}.$$

2.0 Limitations of the Euler turbine theory.

Euler's turbine model in its simplified form is disadvantaged by four assumptions that affect all fan types. These assumptions primarily affect the properties at d_2:

2.1 Assumption of an infinite number of infinitely thin blades.

The assumption of an infinite number of infinitely thin blades implies that slip is zero. Slip is a consequence of finite numbers of blades with finite thickness and has the effect of decreasing the angle of β_2 and increasing the angle α_2 towards a more radial direction. This creates new parameters which change c_2 to c_3, β_2 to β_3 and w_2 to w_3 (see Figure 2 below).

2.2 Shock free flow conditions at entry.

The incoming flow is generally assumed to radial only ($c_{1u} = 0$). This assumption excludes a rotational component unless inlet vanes are fitted. However Eck has stated that on the basis of the established mechanism of energy transfer, some degree of pre-rotation must exist. This would imply shock free entry conditions are not applicable.

2.3 Uniform velocities throughout the process.

The work by Prandtl and Meyer more than 110 years ago had already demonstrated that this condition is unlikely to prevail in any internal flow system where changes in velocity are taking place. Flow visualisation has shown the velocities at the side plate blade leading edge are appreciably higher than those at the back plate leading edge.

2.4 Flow is assumed to be frictionless

The finite blade numbers introduces the effects of friction and boundary layer growth as the flow passes through the impeller blade passages. The work by Prandtl and Meyer more than 110 years ago had already demonstrated that boundary layer growth and friction are inescapable. The growth of the boundary layer within the impeller flow passages acts as a physical obstruction to flow and reduces capacity and increases the risk of stall operation.

A revised form of Euler's turbine model that includes the effects of slip is illustrated below for a centrifugal fan.

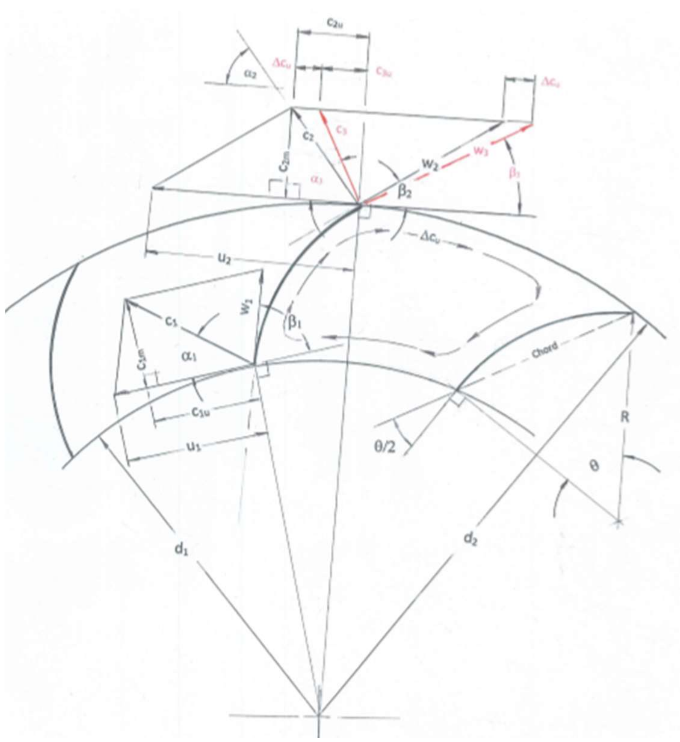


Figure 2 - *Velocity diagram for a centrifugal fan including slip*

For all fan designs, the critical issue is to determine the magnitude of the slip (Δc_u) as this will erode the pressure generated by the fan. This process is illustrated by Figure 2.

Slip is essentially a circulation that that takes place inside the impeller passages and has the effect of reducing the magnitude of c_{2u} to c_{3u}.

3.0 Basic parameters that define a centrifugal fan impeller.

3.1 Blade setting angle.

Blade setting angle is commonly defined in two ways as illustrated in Figure 3 below:

a. Defined as the angle (θ_{LE}) between the blade chord line and a radial line at the blade nose setting circle at d_1.
b. Defined as the angle (θ_{TE}) between the blade chord line and a radial line at the blade trailing edge setting circle at d_2.

In order to calculate the blade setting angle it will be necessary to determine θ, β_2, β_3 and δ_{Stod} as illustrated in Figure 3 below.

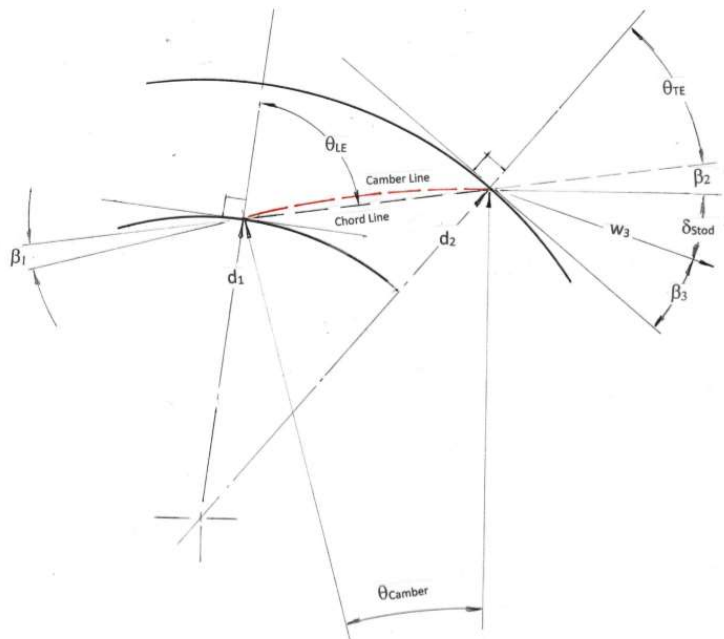


Figure 3 – Blade setting angle

When measured relative to the tangent at d_2, the flow at the blade trailing edge will be tangent to the camber line at d_2 if slip is not included. For a straight blade, this angle will be θ_{TE}.

For a blade with a circular arc camber, the angle will be ($\theta_{TE} + \beta_2$). According to the theory by Stodola, internal recirculation inside the impeller passages will cause the discharge relative flow to deviate by some angle δ_{Stod} which is reflected by Δ_{cu}.

δ_{Stod} is typically between 15^0 and 30^0 depending on blade setting angle and the number of blades.

3.1 Blade width at tip

Based on experience with high specific speed centrifugal fans use in mine ventilation applications, the width of a fan blade at the tip is limited by stress levels and the risk of flow instability. To avoid these problems the blade tip width is generally limited to a maximum of about 0.30D.

3.2 Fan blade profile

The blades in centrifugal fans do not operate as aerofoil wings (as in an axial fans). Centrifugal fans are radial flow machines, and <u>the use of hollow section blades has been driven by the need to maximise stiffness and limit blade bending stresses to manageable levels.</u>

As a general rule, variable thickness aerofoil blades in centrifugal fans are not as efficient as laminar bladed fans. Strength and acoustic benefits govern the use of variable thickness hollow section aerofoil blades.

The choice of blade profile is based on ease of manufacture. The profile thickness is determined by stress considerations.

3.2.1 Fan casing geometry

There are a range of proportions for fan casings that will deliver similar levels of performance. A basic spiral shape has been shown to deliver the highest levels of efficiency.

Minor variations to these proportions <u>do not a have major impact on performance</u>.

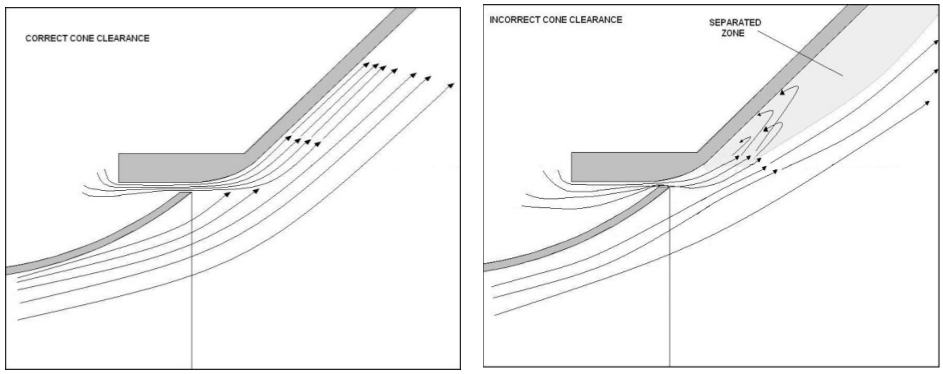
3.5 Stationary inlet cone

The stationary inlet cone is a critically important component that distributes the incoming flow into the impeller inlet. The design process is dominated by the need to ensure that the incoming flow does not separate from the surface of the impeller side plate. If flow does separate from the inner surfaces of the impeller, the separated zone acts as a physical barrier to the flow and at best will significantly degrade fan performance and efficiency. At worst it could result in stall operation and damage to the impeller, fan casings and ductwork.

An inlet cone and impeller assembly are illustrated in Figure 4(a) and 4(b) below.

The flow interactions between the inlet cone and the impeller inlet for any high specific speed design demands an interactive design process between these two elements.

The conical upstream reducer section is designed to accelerate the flow and deliver thin boundary layer and uniform velocity distribution in the flow as it approaches the area of highest velocity at the throat region.



(a) (b)

Figure 4 – Inlet cone/impeller inlet relationships

The features that are needed to achieve these objectives are:

- Precise location of the inlet cone lip inside the impeller inlet ring.
- A toroidal shaped throat turns and decelerates the flow towards the impeller inlet. This process inevitably results in an increase in the boundary layer thickness.
- It is important to ensure that the tangent angle of the flow leaving inlet cone should match the impeller side plate angle as closely as possible (within say 15^0).
- The clearance between the lip of the stationary inlet cone and the impeller inlet ring is controlled within fine limits. This clearance allows for a small amount of high pressure air from the impeller to recirculate back into the incoming flow. The gap and length of this clearance is designed to establish laminar flow which tends to “stick” to the surface it is following (Coanda effect). The laminar flow follows the surfaces of the inlet ring and impeller side plate and re-energises the much thicker boundary layer present in the flow entering from the inlet cone.

4.0 Basic parameters that define a mixed flow fan impeller.

A mixed flow fan impeller is a relatively simple structure that comprises constant thickness circular arc blade profiles that are welded to a steel hub. Once manufactured, the blade setting angle cannot be changed. These fans can be defined by a relatively small number of parameters. The parameters that define the impeller hub are described in Figure 1.5 below:

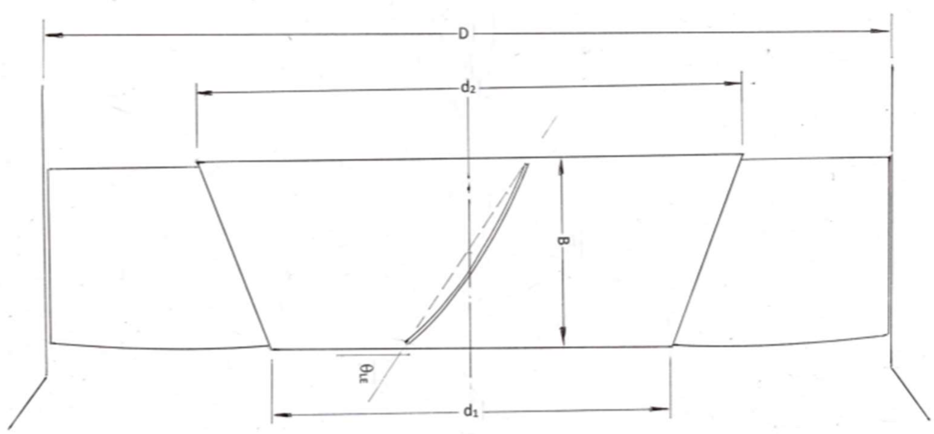


Figure 5 – Geometric relationships

4.1 Blade setting angle.

In common with vane axial fans, the blade setting angle (θ_{LE}) is commonly defined in terms of the angle between the blade tip chord line and the inlet flow or by the angle between the blade tip chord line and plane of rotation.

4.2 Blade chord length and shape.

Blades are generally characterised by relatively small change in chord length along the span and much higher camber angles (typically ranging from 25^0 at the blade tip to 90^0 at the blade root). These values are much higher than would be expected for vane axial fan blades

4.3 Hub ratio.

The mixed flow fan designs involve three diameters as illustrated in Figure 4 below:

- Tip diameter (D)
- Impeller inlet hub diameter (d_1)
- Impeller discharge diameter (d_2)

All three of the above diameters will depend on the required flow and pressure that the fan has to achieve. The defining relationships are d_2/D and d_1/D. The ratio d_2/d_1 reflects the degree of meridional acceleration that is required.

It is important to note that most modern hub designs incorporate profiles that have a spherical or conical shape. The performance is not markedly affected by choice of either hub profile. The width "B" is a consequential dimension that is determined by the blade chord length being considered. While changes in chord length for a given design of known solidity will affect the hub width "B", they do not affect the primary aerodynamic properties for the fan under consideration. The dimension "B" (and the associated slope) is of no relevance provided the diameters d_1 and d_2 are maintained.

4.5 Solidity

Mixed flow fans will operate efficiently over a significant range of solidities. When referenced to the downstream hub diameter (d_2) the blade solidity can vary from 0.7 to 1.8.

4.6 Blade profile.

The blade designs are based on circular arc camber lines and this facilitates the generation of aerodynamically identical blades with substantially different chord lengths.

Kevin J Lownie.
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References/Biliography.

1. Eck, B., *Design and Operation of Centrifugal, Axial Flow and Cross Flow Fans*. Pergamon Press Ltd Oxford (1973). ISBN 0 08 015872 2.
2. Dixon,S.L., *Thermodynamics of Turbomachinery* (3rd Ed). Pergamon Press Ltd Oxford (1978). ISBN 0 08 022722 8
3. Miller, D S., Internal Flow Systems (2nd Ed). BHR Group Limited (1996). ISBN 0 947711 775 5

APPENDIX A Nomenclature and Symbols. The symbols used by Eck are detailed below:

b_1	impeller width at d_1 (m)
b_2	impeller width at d_2 (m)
c_{1u}	peripheral component at d_1 due to pre-rotation imparted by the impeller (m/s)
c_{2u}	peripheral component at d_2 due to rotation of the impeller (m/s)
c_{1m}	meridional component at d_1 (m/s)
c_{2m}	meridional component at d_2 (m/s)
d_1	diameter at inlet to the impeller (m)
d_2	diameter at discharge from the impeller (m)
H	pressure in m of fluid (m) calculated from $\rho gH = \Delta p_{th}$
H_{th}	theoretical pressure rise expressed as m of fluid = $g^{-1}(u_2c_{2u} - u_1c_{1u})$. For radial entry $r_1c_{1u} = 0$
M	moment of momentum = Torque = q x $(r_2c_{2u} - r_1c_{1u})$. For radial entry $r_1c_{1u} = 0$ (Nm)
q	mass flow rate (kg/s)
r	radius (m)
R	blade radius = $c/\{(2*\text{Sin}(\theta/2)\}$
u_1	peripheral velocity of the impeller at d_1 (m/s)
u_2	peripheral velocity of the impeller at d_2 (m/s)
w_1	velocity at d_1 relative to the blade leading edge (m/s)
w_2	velocity at d_2 relative to the blade trailing edge (m/s)
Q	volumetric flow rate (m^3/s)
q	mass flow of fluid (kg/s)
R	Blade radius = $c/(2*\text{Sin}(\theta/2)$
W	power = $M\omega = \omega q(r_2c_{2u} - r_1c_{1u})$. For radial entry $r_1c_{1u} = 0$
α_1	absolute angle of flow (c_1) due to pre-rotation imparted by rotation of the impeller at d_1
α_2	absolute angle of flow (c_2) due to rotation imparted by the impeller at d_2
β_1	angle of the tangent to blade camber line at leading edge relative to the chordal line at d_1*
β_2	angle of the tangent to blade camber line at trailing edge relative to the chordal line at d_2 *
$\Delta p_{th.inf}$	theoretical pressure rise relative to atmosphere for <u>infinite number of blades</u> = ρu_2c_{u2}
$\Delta p_{th.}$	theoretical pressure rise = ρu_2c_{2u} for radial entry (N/m^2) for a number of blades
γ	weight density of m^3 of fluid = ρg (N/m^3)
τ	reaction coefficient
θ	blade camber angle
ρ	mass density of fluid = γ/ρ (kg/m^3)
ω	angular velocity of impeller rotation in Radians/s

*For straight blades $\beta_1 = \beta_2 = 0$