**Centrifugal Fan Design**

In this document is described the design procedure of a centrifugal fan with a radial blades impeller. The input parameters for the design of radial tipped centrifugal fan are the following: Flow Discharge *Q,* Static Suction Pressure *Ps*, Static Delivery Pressure *Pd*, Static Pressure Drop *ΔPS* (Pd – Ps)*,* rotational speed *ω*, Air Density *ρ*, number of blades *Nb,* blade thickness *tb,* Outlet Blade Angle *β2* = 90° and the Suction Temperature *Ts*.

This design procedure is based on the fundamental principles of fluid flow with continuity and energy equations. The design follows the path followed by the air flow from suction to discharge. The procedure by means of iterative steps takes into account the different energy losses along the analysed flow path (hydraulic, leakage and power losses). In the fan the total pressure will rise from the inlet and the outlet section by an amount called *fan total pressure*.

The design method adopted here is a “trial and error” method: in the first step of the design procedure the energy losses are not known, so the fan geometry is obtained by neglecting the flow losses; in the second step the flow losses are estimated by using the first iteration parameters’ values, so the first correction of the previous parameters can be done; in each of the following iterations the parameters correction is estimated. The design process stops when the deviation between the main geometric parameters (i.e. *D*1, *D*2, *ΔP*S) of the ith and the (i-1)th iterations is lower than a fixed threshold.

***Design of inlet duct and impeller***

A conical duct is used to improve the airflow entry inside the impeller and to reduce the energy losses at the impeller inlet: taking into account an optimal good cone contraction (θc < 30) the flow acceleration does not generate any flow separation and noise. To estimate the length of the inlet duct *Lduct* the inlet duct diameter *Dduct* could consider being the 20% higher than the impeller inlet diameter *D1*. Assuming no loss during the 90º turning from duct outlet (also called *impeller eye*) to impeller inlet, the impeller inlet velocity *V1* will have the same absolute velocity of the velocity at the impeller eye. Considering a radial direction for the inlet velocity (*α1* = 90º), inlet meridian velocity *Vm1* iscoincident to the impeller inlet velocity module. Further, let the inlet tip velocity *U*1 (also called peripheral velocity) be higher than the meridian velocity *Vm1* for better induction of flow: *U*1 ­­­­= α *Vm1* with α > 1. This velocity ratio has an important role because a wrong velocity ratio could cause high impeller losses: the simplest assumption can be considered is that the smallest possible entry velocity is required, so that the losses can be reduced (Austin Harris Church, 1944 - Centrifugal pumps).

; (1)

The continuity equation, applied at the inlet section of the impeller, and the relation between the tip velocity and the inlet velocity, allow estimating the inlet diameter *D*1 of the impeller, the tip velocity *U*1 and the inlet velocity *V*1:

(2)

To define the velocity triangle at the impeller inlet, the relative velocity *W*1 as well the angle *β*1 between *W*1 and U1 have to be estimated (add figure of the inlet vel. triangle). The angle *β*1 represents the inlet tip angle and it only depends on the ratio between *V1* and *U1*; this ratio value derives from the hypothesis we made about a better induction flow.

; (3)

The width at impeller inlet, *B1*, is estimated by using the continuity equation at this section:

(4)

In order to define the impeller outlet the fan power *Pimp* has to be estimated by using the flow input parameters and considering a 10% extra of power to consider flow recirculation and impeller exit hydraulic losses. This work is proportional to the specific work done *Ws* by the mass flow rate:

; (5)

To estimate the velocity triangles at the impeller exit and so the outer diameter of the impeller the Euler equation on a generic blade vane between the inlet and the outlet sections of the impeller (section 1-2) can be applied (add picture here). To do that the fluid slip phenomena has to be taken into consideration: the fluid slip is the deviation in the angle at which the fluid leaves the impeller from the impeller's blade/vane angle. So, an estimation of this phenomenon is useful in determining a more accurate value work input between the impeller and the fluid and the velocity triangles at the impeller exit. Many numerical and experimental studies stated that this phenomenon is directly linked to the number of the impeller blades: theoretically, to get the perfect ideal flow guidance, one can infinitesimally increase the number of thin vanes so that the flow should leave the impeller at an exact vane angle, on the other hand this means an increasing of the energy losses inside the impeller. A measure of this phenomenon is the *slip factor*: it generally increases with the increase in the number of impeller blades, thus, accounts for one of the important parameter for losses. Esperimental observations showed as the slip factor also depends on other parameters, such as the outlet blade angle *β*2, the ratio between the relative velocity and the tip velocity at the blade outlet *φ2* (*W2/U2*)and the diamters ratio D2/D1. In eq (6) are reported some numerical espressions of the slip factor σ proposed by some authors (Stodola, Stanitz, Balje and Wiesner).

; (6)

The fluid slip influence has been estimated first by using the *Stanitz* formula in eq. 6 for *β*2 = 90º then the complete formula has been considered to improve this factor estimation. The component of the velocity *V2* along the tangential direction *V2t* is affected by the slip factor, *V2t* is lower than the tip velocity module *U2* of the slip factor:

(7)

Once the velocity correction has been estimated the Euler power *PE* can be evaluated and the outer diameter of the impeller *D2* as well:

; (8)

The meridian component of the outlet velocity *V2m* ­can be estimated by applying the continuity equation at the out vane section considering that the impeller width is the same as the inlet (*B1* = *B2*). Once the outlet velocity is deviated by the fluid slip, the real outlet blade angle *β*2 is not 90º anymore (*β2*­’ < 90º) and the tangential component of the relative velocity at the impeller outlet *W2t*, is not zero because of the fluid slip, but is proportional to the tip velocity by the slip factor.

(9)

***Design of the volute casing***

To design the volute casing, the velocity at the volute outlet section *V4* and the width of this section have to be estimated. In order to do that the Bernoulli equation is applied between the inlet section of the impeller and the outlet section of the fan casing, and the continuity equation at the outlet section of the casing.

(10)

Where *Ws* is the specific work done inside the impeller, previously estimated by eq. 5, *R3* is the volute radius at the impeller outlet section (add figure of the fan volute), *clv* is the clearance between the volute and the impeller outlet (*clv* = 1%·*D2*) and *B4* is the volute outlet width (usually 2-3 *B1*).

The shape of the volute has a linear variation with the radius starting from *R3* up to *R4­*.

***Design of the blade***

The blade profile is an arch of the circumference of radius *Rb*: the value of this radius can be estimated because the side tangents are known (*β*1 and *β*2 previously estimated).

(11)

***The correction factors***

After obtaining a preliminary dimension of the fan further iterations are needed in order to get the optimum design parameters. To estimate the optimum parameters set an iterative estimation of the hydraulic, leakage and power losses at different flow passages is performed: this correction is obtained by using the previous iteration design point discharge and pressure head across the impeller stage as given in input data.

*Leakage losses at impeller inlet*

(12)

where *Cd* is the discharge coefficient (generally around 0.6-0.7), *Ps* is the parameter proportional to the static pressure drop (2/3 *ΔPs*) and *cl* is the clearance between the duct and the impeller inlet (1% of *D*1).

*Suction pressure loss*

(13)

where *ki* is the loss factor, generally assumed to be in the range 0.5 – 0.8.

*Impeller pressure loss*

(14)

where *kii* is the loss factor assumed to be in the range 0.2 – 0.3 in the case of sheet blade, *kii* < 0.2 for a airfoil blade profile.

*Volute pressure loss*

(15)

where *kiii* is the loss factor assumed to be around 0.4.

*Discharge and pressure corrections - Efficiency estimation*

At the second iteration the new input parameters for the fan design and are assumed to be as corrections of the previous iteration parameters, respectively, and , by means of the leakeage loss and the pressure losses estimated in the previous step.

(16)

The efficiency of the designed turbine can be expressed as the ratio between the previous and the last step of design.

; (17)

In order to design the shaft diameter the ideal torque has to be estimated. First the disk friction loss is computed as follows:

(18)

where C*f* is the friction factor that can be estimated by using the kinematic and geometric condition of the flow (this coefficent can be considerd of the order of 0.005, but a rough estimation is suggested). The friction loss allows to estimate the power loss *Pdf* and so the ideal power *Pideal* and the ideal torsion *Tideal*.

(19)

The ideal diameter of the fan shaft *Dshaft* is estimated as follow:

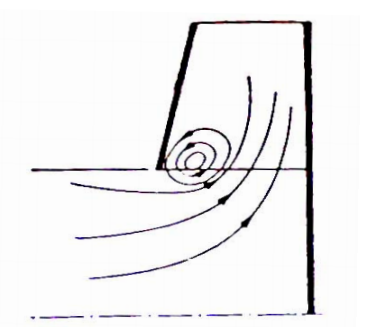
(20)

Where *ft*is the a safety factor (4) and *τ* is the yield strength of the steel (Steel 1090 mild *τ* = 247 MPa;Steel AISI 302 *τ* = 520 MPa).

**Compressible flow and variable blade width**

***Design of inlet duct and impeller***

The factors involved in determining the size of the axial breadth B1 of a blade entry can be readily obtained. Before the introduction of air into the impeller, the air must be turned through an angle of 90° (approximately) from the axis of the suction of the intake duct. This is analogous to a change of direction occurring at a bend. The inside radius of curvature, however, is not always adequate in fans because of the insufficient room. The rounding-off of the front shroud is often omitted to reduce the manufacturing costs. Even due to a small radius, the separation flow occurs, and it will lead to pressure loss which will indirectly influence the impeller performance. A secondary effect is a “backflow” which could arise in the remaining portion of the impeller entrance. The air flowing back into the suction or intake region is often responsible for the large loss of energy. To avoid this influence upon the impeller, flow separation must be prevented. The most effective way to prevent this effect is to better guide the flow inside the impeller vanes and give some acceleration to the flow (B1 = 0.21 ·D1).



**Fig. Xx -** Backflow at the inlet of impeller due to large inlet breadth B1

The variation of the temperature affects the density and pressure air flowing inside the impeller. The density of air can be estimated by using the state equation of gas:

(21)

Once the inlet velocity V1 and the inner diameter D1 are estimated by using Eqs. 1-3, so the equation of gas can be used to estimate the absolute static pressure at the impeller inlet:

(22)

A different pressure between the air inside the impeller and the atmospheric one affect the air temperature and the density:

(23)

The new value of temperature and pressure affect the air density; the new value of the density ρ1 can be estimated by using Eq. 21. Also, the flow rate changed due to the temperature variation inside the impeller: the new flow rate value is given by the product of the initial flow rate times the ratio between the atmospheric density over the air density inside the impeller inlet.

(24)