MAXIMUM ACHIEVABLE EFFICIENCY OF CENTRIFUGAL FANS WITHOUT HOUSING

Yingan XIA[1], Şaban ÇAĞLAR[2], Heiko RATTER[2], Martin GABI[2]

[1] *Punker GmbH, Niewark 1,24340 Eckerfoerde, Germany*

[2] *FachgebietStroemungsmaschinen, Karlsruhe Institut of Technology (KIT), Kaiserstraße 12, 76131 Karlsruhe, Germany*

SUMMARY

The present paper is focused on centrifugal backward curved fan impellers (excluding motor). Applying the Euler's equation to an idealized flow in the fan impeller, the relation between the efficiency $\eta_r$of impelleras to ErP Directive [1] and the hydraulic efficiency of the blading η_{hydr} is derived. Thus the maximum efficiency η_r of an aerodynamically "lossfree" fan impeller can be estimated by setting $\eta_{hydr} = 1$. Inversely on the basis of testing result η_r of a fan impeller, the theoretic, hydraulic efficiency η_{hydr} can be determined as well enabling to evaluate the potential for further improvement. Comparing three sample impellers with different blade geometries shows that the presently achieved fan efficiency of $\eta_r = 73\%$ stands for a remarkably high level. The potential for further aerodynamic improvement is highly limited.

INTRODUCTION

The new ecodesign directive for energy related products (ErP) of the EU [1] puts a strong claim on efficient fans.After 2013 fans with electric power consumption at maximum efficiency between 125W and 500kW can only be sold and operated, if the prescriptive limits of efficiency are fulfilled. In 2015 the minimum efficiency rates are increased even more. Against this background a number of efforts in research and development of fan industry is scoped on the increase of efficiency of the complete fan. The improvements have to be done for all parts of the fan, including impeller, motor, driving mechanism and electronic control.

The scope of this project is the impeller of a radial fan with backward curved blades. This type of fan can operate without a spiral casing and is a typical design for fan systems like HVAC, AHU with increasing popularity. The main advantages of this type are the low requirements of spacing and the direct drive mechanism. On the other hand the efficiency of a direct drive fan is a little lower due to the lack of the spiral housing, which converts the dynamic energy of the air into static pressure. In this case, the dynamic pressure at the outlet of the impeller has to be treated as loss.

MOTIVATION

Maximum efficiency rates of radial fans without spiral casing can be found in different industrial brochure. Depending on fan power, they vary for instance for fans up to 1KW electrical power input between 65% and 76% [2]. The question is how these values can be evaluated. It is also important to know the physical limit in terms of efficiency rates for a radial fan (impeller), from the fluid dynamical point of view.The information then can be taken as a sort of reference for setting values within ErP to make the efficiency target high, reasonable and realistic.

This paper shows the approach to calculate the maximum achievable efficiency rate as a function of the operating point (volume flow V and pressure increase Δp) and the given geometry data (diameter, meridianshape) for an idealized flow. The calculated value shows the maximum achievable efficiency and can be taken into account for further developments and political guidelines like ErP.

Characteristic flow of a fan without spiral casing

Fig. 1 shows the path of the fluid flow within a radial fan with a rotating diffusor at the outlet of the impeller. The air is moving from the environment "E", passing the inlet cone (0). The mechanical energy P_W input through the driving shaft (W) will be converted intoflow energy between the leading (1) and trailing edge (2) of the blade. The air leaves the impeller at (3).

This approach analyzes the flow from (E) to (3). Velocities and velocity components are defined as usual as shown in Fig. 2.

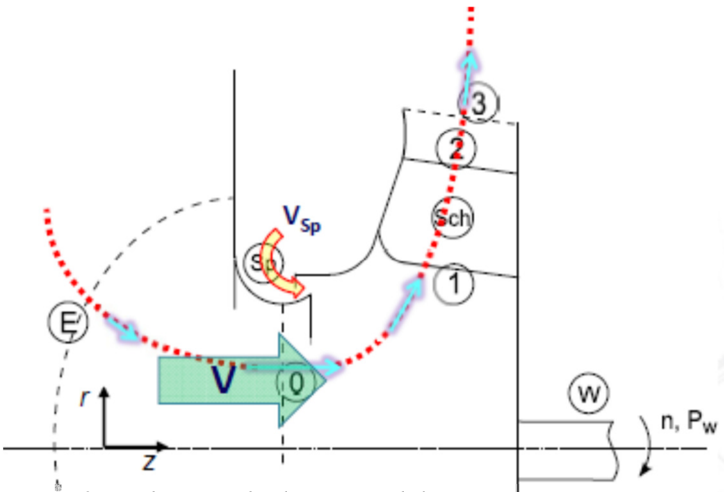


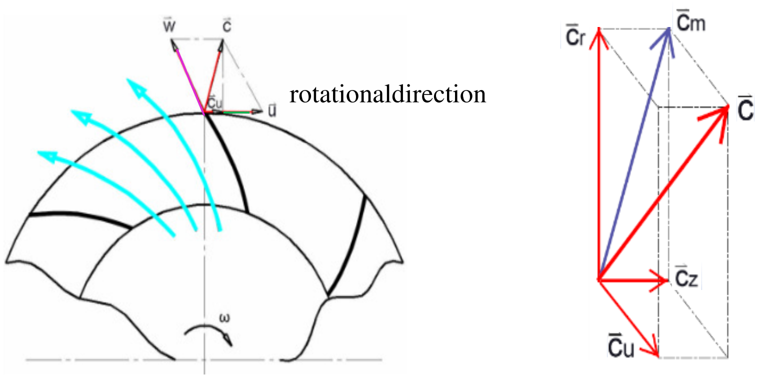
Fig. 2 Definition of velocities and their components

Indices:

E: Entrance to the fan ($c_E \approx 0$)
Direction of fluid flow: 0; 1; 2; 3
Sch: Blade between leading 1 and trailing edge 2
Sp: Gap between inlet cone and hub
W: Driving shaft

Following assumptions are made for an ideal flow:

- Flow is steady in time ($\frac{\delta}{\delta t} = 0$);
- Fluid is incompressible (ρ = const.);
- Flow can be characterized by averaged values of the meridianvelocity c_m and the circumferential velocity c_u.
- Inflow upstream of the blade is swirl free ($c_{1u} = 0$) and swirl after the blade is constant ($r_3 \cdot c_{3u} = r_2 \cdot c_{2u}$).
- Flow between (E) and (0) is without loss ($p_{ges.0} = p_{ges.E}$).



u: rotation velocity of the impeller
w: relative velocity
c: absolute velocity
c_u : circumferential component of c
c_m: meridian component of c
c_z: axial component of c
c_r: radial component of c

Fig. 2 Definition of velocities and their components

The energy transport of the ideal fluid flow within the impeller can be calculated by the Eulerian equation [3]. Consequently the maximum achievable additional energy in the fluid is P_{Sch}.

$$P_{Sch} = \rho \cdot V_{Sch} \cdot (u_2 \cdot c_{u2} - u_1 \cdot c_{u1}) \quad (1)$$

with $$P_{Sch} = P_W \cdot \eta_{mech} \cdot \eta_{rsr} \quad (2)$$

and $$V_{Sch} = V + V_{Sp} = \frac{V}{\eta_{vol}} \quad (3)$$

with
ρ air density
V_{Sch} sum of the main volume flow V and the volume flow through the gap V_{Sp}
P_W mechanical power at the shaft
η_{mech} mechanical efficiency of driving unit (incl. bearings etc.)
η_{rsr} efficiency due to the friction losses at the outer parts of the impeller
η_{vol} volumetric efficiency due to gap flow

The maximum achievable fluid energy P_{Sch} can also be written as a function of the pressure difference Δp_{sch}.

$$P_{Sch} = \Delta p_{sch} \cdot V_{Sch} \quad (4)$$

with Δp_{sch} maximum total pressure difference

On the other hand the total pressure difference can be split into the components of the usable pressure difference Δp_f and the kinetic energy at the outlet of the impeller $\rho \cdot \frac{(c_3)^2}{2}$.

$$\Delta p_{ges} = \Delta p_f + \rho \cdot \frac{(c_3)^2}{2} \quad (5)$$

with Δp_f usable fluid energy $(p_3 - p_{tot.E})$ and $c_3{}^2 = c_{3m}{}^2 + c_{3u}{}^2$

Hydraulic efficiency η_{hydr}

The hydraulic efficiency η_{hydr} quantifies the ratio of the total pressure difference of the fluid Δp_{ges} in relation to the maximum possible total pressure difference Δp_{Sch}

$$\eta_{hydr} = \frac{\Delta p_{ges}}{\Delta p_{Sch}} \tag{6}$$

eq. 5 and eq.6 lead to:

$$\eta_{hydr} \cdot \Delta p_{Sch} = \Delta p_f + \frac{\rho \cdot c_{3m}{}^2}{2} + \frac{\rho \cdot c_{3u}{}^2}{2} \tag{7}$$

With $c_{3m} = \frac{V_{Sch}}{A_3}$ and $A_3 = \pi \cdot D_3 \cdot b_3$

whereas A cross section of the blade channel,
D diameter of the impeller,
b width of the blade.

The left hand side of eq.7 shows the total achieved fluid energy, whereas the right hand side shows its parts of pressure and kinetic energy. The following calculations, reckoning the components of kinetic energy, make use of the squares of its mean values; a proper way would be in using the mass-flow weighted mean values.

"Free-blowing" efficiency η_f

The "free-blowing" efficiency is the quantity for the usable part of the fluid energy delivered by the fluid machine.

$$\eta_f = \Delta p_f \cdot \frac{V}{P_W} \tag{8}$$

Equations 2, 3, 8 and 4 serve for the maximum total pressure difference

$$\Delta p_{Sch} = P_W \cdot \eta_{mech} \cdot \frac{\eta_{rsr}}{\left(\frac{V}{\eta_{vol}}\right)} = \frac{\Delta p_f}{\eta_f} \cdot \eta_{mech} \cdot \eta_{rsr} \cdot \eta_{vol} \cdot \tag{9}$$

Respectively in the following dimensionless form:

$$\Psi_{Sch} = \frac{\psi_f}{\eta_f} \cdot \eta_{mech} \cdot \eta_{rsr} \cdot \eta_{vol} \tag{10}$$

The dimensionless numbers are defined as follow:

fluid flow $\varphi = \frac{V}{A_2 \cdot u_2}$ cross section: $A_2 = \pi \cdot \frac{D_2{}^2}{4}$

pressure difference $\psi = \frac{\Delta p}{\frac{\rho \cdot u_2{}^2}{2}}$ rotation velocity: $u_2 = \pi * D_2 \cdot n$

power $\lambda = \frac{P}{\frac{A_2 * \rho * u_2{}^3}{2}}$ number of revolutions: n

The conservation of mass and the proposition of a constant swirl lead to:

$$c_{3m} = \frac{\varphi}{\eta_{vol}} \cdot \frac{A_2}{A_3} \cdot u_2 \tag{11}$$

$$c_{3u} = c_{2u} \cdot \frac{D_2}{D_3} \tag{12}$$

The swirl free inflow condition and eq. 1, 4 and 12 lead to

$$c_{3u} = \frac{\Delta p_{Sch}}{\rho \cdot u_2} \cdot \frac{D_2}{D_3} = \frac{\Psi_{Sch}}{2} \cdot \frac{D_2}{D_3} \cdot u_2 \qquad (13)$$

For simplification, the mean value of the (mass-flow weighted) product of $u_2 \cdot c_{2u}$ in eq. 1 is replaced by the product of u_2 and an appropriate mean value of c_{2u}.

The hydraulic efficiency can be calculated by eq. 11, 13 and 7 as a function of the operating point and the geometric parameters of the impeller in the following dimensionless form:

$$\eta_{hydr} \cdot \psi_{Sch} = \psi_f + \left[\frac{\varphi}{\eta_{vol}} \cdot \frac{A_2}{A_3}\right]^2 + \left[\frac{\psi_{Sch}}{2} \cdot \frac{D_2}{D_3}\right]^2 \qquad (14)$$

Volumetric efficiency η_{vol}

The volumetric efficiency of eq. 14 is a function of the main volume flow V and the volume flow through the gap V_{Sp}, and is defined below:

$$\eta_{vol} = \frac{V}{V+V_{Sp}} = \frac{1}{1+\frac{V_{Sp}}{V}} \qquad (15)$$

The volume flow through the gap V_{Sp} can approximately be calculated as follows:

$$V_{Sp} = \alpha_{Sp} \cdot A_{Sp} \cdot \sqrt{\frac{2 \cdot \Delta p_{Sp}}{\rho}} \qquad (16)$$

with

the cross section area:	$A_{Sp} = \pi \cdot s \cdot D_{Sp}$
width of the gap:	s
static pressure difference at the gap:	$\Delta p_{Sp} = p_3 - p_0 = \Delta p_f + \rho \cdot \frac{c_0^2}{2}$
gap volume flow coefficient:	$\alpha_{Sp} = 0.75$

The volumetric efficiency η_{vol} is a function of the ratio $\frac{V_{Sp}}{V}$, whereas:

$$\frac{V_{Sp}}{V} = \alpha_{SP} \cdot A_{Sp} \cdot \frac{\sqrt{\frac{2 \cdot \Delta p_{Sp}}{\rho}}}{\varphi \cdot A_2 \cdot u_2} \qquad (17)$$

with $$\Delta p_{Sp} = \Delta p_f + \rho \cdot \frac{c_0^2}{2} = \left[\Psi_f + \varphi^2 \cdot \left(\frac{A_2}{A_0}\right)^2\right] \cdot \frac{\rho \cdot u_2^2}{2} \qquad (18)$$

The volumetric efficiency η_{vol} can be calculated:

$$\eta_{vol} = \frac{1}{\left\{1+\alpha_{SP} \cdot \frac{A_{Sp}}{A_2} \sqrt{\frac{\Psi_f}{\varphi^2} + \left(\frac{A_2}{A_0}\right)^2}\right\}} \qquad (19)$$

Estimation of η_f

Combining eq. 10 with eq. 14 and multiplying both sides with ${\eta_f}^2$, a quadratic equation of η_f as a function of the dimensionless operating point (φ, ψ_f) and geometric parameters (D, A) result in:

$$\boldsymbol{\eta_f}^2 \cdot \left[\psi_f + \left(\frac{\varphi}{\eta_{vol}} \cdot \frac{A_2}{A_3}\right)^2\right] - \boldsymbol{\eta_f} \cdot \eta_{hydr} \cdot \Psi_f \cdot \eta_{mech} \cdot \eta_{rsr} \cdot \eta_{vol} + \left(\frac{\Psi_f}{2} \cdot \eta_{mech} \cdot \eta_{rsr} \cdot \eta_{vol} \cdot \frac{D_2}{D_3}\right)^2 = 0 \qquad (20)$$

This equation has two solutions:

$$\eta_f = \frac{Q}{2 \cdot R} \cdot \left\{ \eta_{hydr} \pm \sqrt{{\eta_{hydr}}^2 - R \cdot \left(\frac{D_2}{D_3}\right)^2} \right\} \tag{21}$$

with $R = \Psi_f + \left(\frac{\varphi}{\eta_{vol}} \cdot \frac{A_2}{A_3}\right)^2$ and $Q = \psi_f \cdot \eta_{mech} \cdot \eta_{rsr} \cdot \eta_{vol}$

Only the solution with ${\eta_{hydr}}^2 \geq \left\{ \psi_f + \left(\frac{\varphi}{\eta_{vol}} \cdot \frac{A_2}{A_3}\right)^2 \right\} \cdot \left(\frac{D_2}{D_3}\right)^2$ is possible, and only the solution of the summation is of interest.

Consequently the theoretical solution for the efficiency η_f is:

$$\eta_f = \frac{Q}{(2 \cdot R)} \cdot \left\{ \eta_{hydr} + \sqrt{{\eta_{hydr}}^2 - R \cdot \left(\frac{D_2}{D_3}\right)^2} \right\} \tag{22}$$

The efficiencies η_{mech} and η_{rsr} in eq. 22 are negligible for radial fans without spiral casing. The volumetric efficiency can be calculated as shown in eq.19. Therefore the “free-blowing” efficiency can be calculated with eq. 22.

In case there is no rotating diffusor between the trailing edge and the outlet of the impeller, the diameters D_2 and D_3 will be equal and $\frac{D_2}{D_3} = 1$.

Mathematically eq. 20 can be used for determining the best available η_f in relation to a given hydraulic efficiency η_{hydr}, the desired operating range (φ, Ψ_f) and the geometric parameters of the impeller (D_3, A_3). For a real technical problem (V, Δp_f), however, the maximization of η_{hydr} requires a reasonable combination between the diameter of impeller D_2 and the operating speed n, which should in our opinion be chosen in a strongly limited range. Thus, the dimensionless parameters φ, Ψ_f are actually more or less fixed.

The volumetric efficiency η_{vol} can substantially be optimized through the inlet cone.

Thus, eq. 20 merely contains D_3, A_3 as free parameters having influence on η_f. In order to maximize η_f, the speed out of the impeller c_3 should as much reduced as possible, meaning to choose as large D_3, A_3 as possible. There is indeed an upper limit from the aero dynamical and structural point of view as well.

Eq. 20 might be an important aid to optimize η_f, but the impeller-"designer" has to take into account of other issues like construction limitations and costs.

Analysis of three different radial impellers

Fig. 3 shows three different radial fans from the product brochure of an impeller manufacturer, which have the same diameter of the blades and comparable width of the blades, however, strongly different blade geometries (ref. to table 1). Impeller 3 additionally has a rotating diffusor.

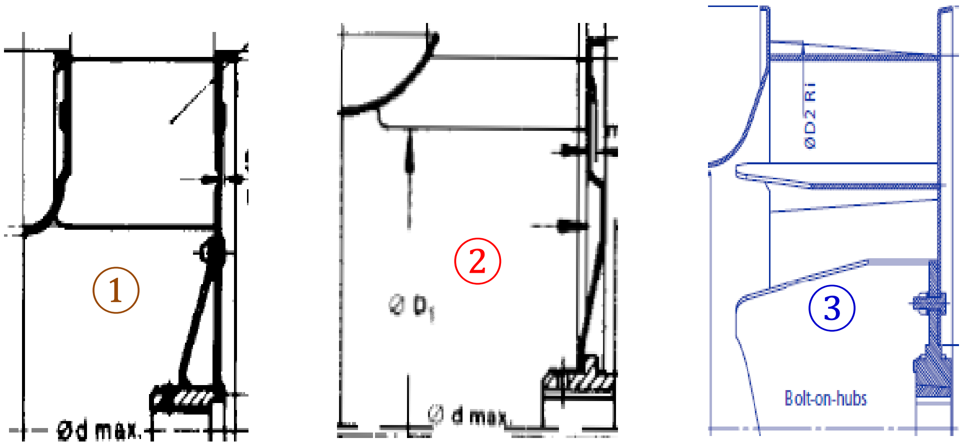


Fig. 3 Three radial fan impellers with different geometries

Table 1 Main dimensions of the three fan impellers

fan impeller	D_2	D_3	D_{Sp}	b_2	A_0	A_2/A_3
	m	m	m	m	m^2	-
①	0.400	0.400	0.209	0.106	0.035	1.000
②	0.400	0.400	0.289	0.112	0.067	1.000
③	0.400	0.450	0.280	0.125	0.062	0.907

All these fans were performed without housing at a testing rig accordant to ISO 5801, Cat. A. Fig. 4 shows the characteristic curves in dimensionless form as defined in eq. 10.

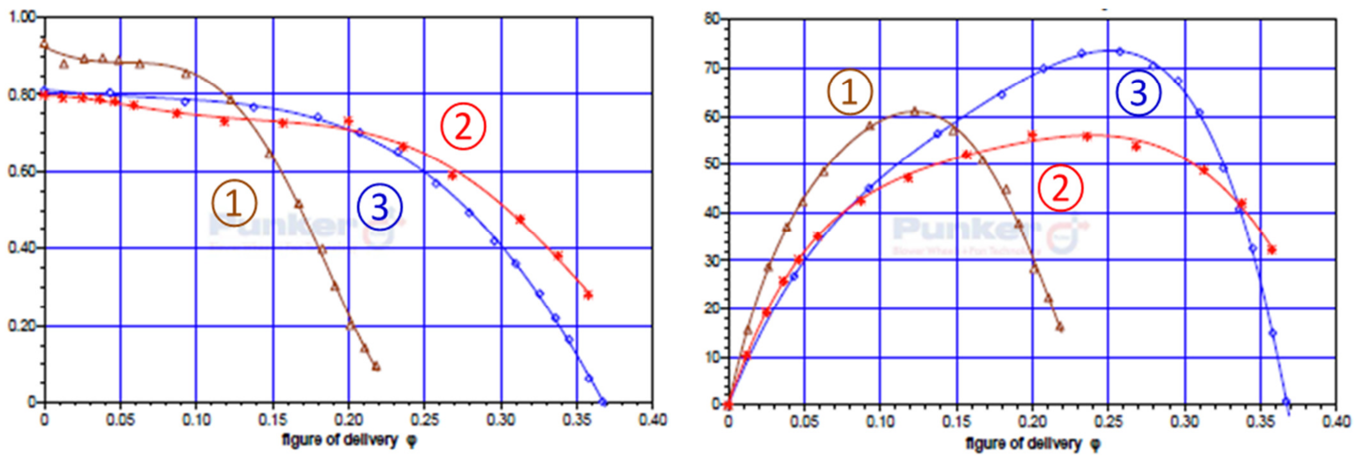


Fig. 4 Dimensionless characteristics, left: pressure number ψ_f; right: efficiency η_f

Estimation of η_{hydr} based on real performance test result of a fan

The hydraulic efficiency η_{hydr} can be estimated by means of eq. 14 on the base of performance data of a fan. For this analysis, only the operating point with a maximum efficiency is of importance, because only at this point, the flow through the impeller can approximately be treated as ideal.

The theoretical hydraulic efficiencies of the mentioned three fan impellers appear in table 2.

Table 2 The really achieved efficiency $\eta_f$and theoretically expectable maximum hydraulic efficiency η_{hydr}

fan impeller	φ	ψ_f	$\eta_{f\_Messung}$	$\eta_{mech} \cdot \eta_{rsr}$	α_{Sp}	η_{vol}		η_{hydr}
①	0.123	0.787	61.0%	99%	0.75	94.0%		95.9%
②	0.236	0.666	56.0%	99%	0.75	95.9%	⇨	90.5%
③	0.257	0.568	73.3%	99%	0.75	96.5%		94.6%

The different operating points (φ, ψ_f) and different efficiencies η_f, which fallbetween 56% and 73%, lead to different hydraulic efficiencies, giving a hint at a theoretical maximum of an improvement potential.

In case a new impeller has to be developed for achieving a specified duty point (φ, ψ_f), you can then choose different geometric parameters and estimate in advance the theoretical hydraulic efficiency η_{hydr} accordingly, without knowing the details of blade geometries. This could be a good indicator to evaluate the potential for improving the efficiency of the impeller.

Estimation of the maximum achievable "free-blowing" efficiency η_{f_th}

With the assumption of a hydraulic efficiency of η_{hydr}=1, the maximum achievable "free-blowing" efficiency η_{f_th} can be calculated by using eq. 22. Table 3 shows this approach for the above three impellers. The difference between the maximum theoretical achievable efficiency η_{f_th} and the real measured efficiency η_f varies from 4% to and 12%. The impeller 3 with 73% of η_f has to be considered as a remarkably optimized product from aero dynamical point of view, because the theoretical potential for an improvement is highly limited.

Table 3 the theoretical limit of maximum achievable efficiency of fan impeller without housing η_{f_th}

fan impeller	φ	ψ_f	η_{hydr}	$\eta_{mech} \cdot \eta_{rsr}$	α_{Sp}	η_{vol}		η_{f_th}	$\Delta\eta_f$
①	0.123	0.787	100%	99%	0.75	94.0%		66.0%	5.0%
②	0.236	0.666	100%	99%	0.75	95.9%	⇨	67.9%	11.9%
③	0.257	0.568	100%	99%	0.75	96.5%		76.8%	3.5%

Potential of increasing the efficiency

Two main aspects have to be carried outat the comparison of the three impellers:

- The theoretical maximum achievable efficiency η_{f_th} is a function of the specified operating points and the given geometric parameters of the impeller. Without housing η_{f_th} will always be limited due to the fact, that the kinetic energy at the outlet (resp. the velocity components c_{3m} and c_{3u}) has to be treated as loss.

- The measured „free-blowing" efficiency η_f can more or less differ from the theoretically maximum achievable efficiency η_{f_th}, significantly depending on the geometry of the impeller (mainly the blade geometry). An aerodynamically well shaped geometry will lead to a small difference between η_f and η_{f_th}.

In order to increase the fan efficiency, two active measures have to be taken. The main focus should be lay on the aero dynamical optimization of the impeller. Secondly the efficiency can also be improved in whatever a way to convert the kinetic energy out of the impeller into usable static energy e.g. by using a guide vane. From the practical point of view, the focus should also be lay on the optimization of the complete fan, even better on an optimum matching between the fan and operation system. Beside impeller, other components like motor, electronic control etc. have to operate efficiently as well.

The Impeller 3 with a rotating diffusor enables a maximum efficiency of 73%. Combined with an efficient motor, it can easily fulfill the target efficiency η_{target} as to the ecodesign directive Lot11, also for step 2 from Jan. 2015 on.

CONCLUSION

In order to get in line with the new ecodesign directive, the efficiency of a number of existing fans has to be improved. By means of the equations derived in this paper, the potential for an efficiency improvement of centrifugal backward curved fans can be estimated on basis of the expected operating point and the geometry of impeller. The analysis of three existing wheels shows that the presently achieved static efficiency of the wheel $\eta_r = 73\%$ represents a remarkably high level. The potential for further aerodynamic improvement is highly limited.

BIBLIOGRAPHY

[1] EU regulation No.327/2011, *implementing Directiv 2009/125/EC of the European Parliament of the Council with regard to ecodesign requirements for fans driven by motors with an electric input power between 125W and 500kW*, 30 March **2011**

[2] cci Zeitung, (*German specialjournal, www.cci-dialog.de*), Page 16, 08/**2011**

[3] Ackert, J. *Eulers Arbeiten über Turbinen und Pumpen*. Sonderdruck. Zürich: Orell Füssli, **1957**