INVESTIGATION ON THE MAXIMUM STATIC EFFICIENCY OF AXIAL FANS

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SUMMARY

The range of kinetic energy at the outlet of a fan depends on the fan design and the operation point. In most configurations the kinetic energy at the outlet is a loss and thus affects the fan efficiency. Diffusers can be installed to reduce these losses. In this study the maximum static efficiency related to the total efficiency η_{fs}/η_f is analyzed for a fan with and without outlet guide vanes. In order to achieve a further efficiency increase, the effect of a diffuser is studied. The kinetic energy can be divided into two components, a meridional and a circumferential component. Both components depend on the hub-to-tip ratio and the swirl distribution. In this study the common free vortex design is considered and the hub-to-tip ratio is chosen according to the Strscheletzky criteria [1]. In the first part, the maximum static efficiency of fans with and without outlet guide vanes is analyzed. The outlet guide vanes lead to the elimination of swirl and thus increase the efficiency. A further possibility to increase the efficiency is the installation of an annular diffuser downstream the axial fan. The diffuser reduces the kinetic energy at the outlet and thus the Carnot losses. A large reduction of the kinetic energy is achieved by a big area ratio of the diffuser. Since the opening angle of the diffuser is limited this can lead to long components. There is a correlation between area ratio, length and losses in the diffuser. The static efficiency is analyzed for a diffuser length which corresponds to the outer fan diameter. There is a significant increase compared to the latter configurations.

INTRODUCTION

Fans are used to provide a desired volumeflow q_V in a system at a certain pressure height p. If the fan is installed at the end of the system, the kinetic energy at the outlet of the fan is a loss. However, even if the fan is located at the inlet of the system or inline, the kinetic energy can often not be recovered.

The static pressure difference p_{fs} (eq.2) is defined as the total pressure head p_f of the fan (eq.1) minus the dynamic pressure p_{d2} at the outlet. The nomenclature in this study is chosen according to the standard *DIN EN ISO 5801:2014-12, Fans - Performance testing using standardized airways*. If velocity v_2 at the outlet is higher than $65\ m/s$ the stagnation pressure p_{sg} has to be used instead of the total pressure p_{tot}.

$$p_f = p_{tot,2} - p_{tot,1} \tag{1}$$

$$p_{fs} = p_f - p_{d2} \tag{2}$$

In 2011 the European commission has established the Commission Regulation 327/2011 [2] where minimum efficiency for fans with electrical motors are defined. Since the electrical components are not addressed in this study, the shaft power P_{shaft} of the motor is considered. The efficiency η is defined as

$$\eta = \frac{p\, q_V}{P_{shaft}} \tag{3}$$

The maximum achievable static efficiency η_{fs} of the fan depends on the design point of the fan and constructional details. The goal of this study is the estimation of the maximum static efficiency η_{fs} related to the total efficiency η_f dependent on the design point. A distinction is made between fans with, respectively without outlet guide vanes OGV and fans with OGV and diffuser.

In order to evaluate the efficiency, the pressure rise p and the volume flow q_V are expressed in a non-dimensional way (eq.4, eq.5). The outer fan diameter D, the rotational speed n and the air density ρ are used as reference values.

$$\text{Volume number} \qquad \varphi = \frac{q_V}{\frac{\pi^2}{4}\, D^3\, n} \tag{4}$$

$$\text{Pressure number} \qquad \Psi = \frac{p}{\rho\, \frac{\pi^2}{2}\, D^2\, n^2} \tag{5}$$

There is a relation of η_{fs} to η_f and the corresponding pressure numbers. This can be expressed as

$$\frac{\eta_{fs}}{\eta_f} = \frac{\Psi_f - \Psi_{d2}}{\Psi_f}. \tag{6}$$

The ratio of the static to total efficiency is only affected by the outlet losses. Therefore the goal is to minimize Ψ_{d2}.

The velocity at the fan outlet c_2 is split in two components, the meridional velocity c_{m2} and the circumferential velocity c_{u2}. In order to determine the dynamic pressure p_{d2}, the energetic average of the velocity at the outlet is computed. The radius r is replaced by the non-dimensional formulation $r^* = 2r/D$. The hub-to-tip ratio ν is introduced as D_{hub}/D. The corresponding non-dimensional pressure number Ψ_{d2} for a rotationally symmetric flow field is defined as

$$\Psi_{d,2} = \frac{1}{\pi^2\, D^2\, n^2\, \dot{m}} \int_{\nu}^{1} \frac{1}{2}\, \left(c_{m2}(r^*)^2 + c_{u2}(r^*)^2\right)\, c_{m2}\, \rho\, 2\pi r^* dr^* \tag{7}$$

The installation of outlet guide vanes can strongly reduce the kinetic energy at the outlet. Outlet guide vanes convert the swirl (c_u) in a further pressure rise. Diffusers reduce the mean flow velocity and therefore the outlet losses.

In order to determine Ψ_{d2} and thus the efficiency η_{fs} to η_f, information about the velocity profile and the hub-to-tip ratio ν must be provided. In Horlocks textbook Axial Flow Compressors [3] different vortex design methods are discussed and analyzed. The free-vortex design has been very common in the past for low pressure axial fans and still today. Marcinowski [4] uses this load distribution and the Strscheletzky criterion [1] for the hub-to-tip ratio to find an optimal fan design. The idea of this hub criterion is to use a hub with the same size as the forced vortex core in a swirling flow. In Carolus textbook Ventilatoren [5] different criterions for the hub design are summarized. A more recent study on the optimization of low pressure fans was published by Bamberger [6]. He used CFD trained artificial networks to increase the static efficiency by optimizing the load distribution and hub-to-tip ratio. This way he could realise an increase of the efficiency, compared to classical design methods.

METHODOLOGY

The profile parameters α_m and α_u are introduced to analyze the flow profile at the outlet. These parameters give the deviation of the energetic profile average to a reference profile. The non-dimensional quantities φ, Ψ, ν are used as references.

$$\alpha_m = \frac{\left(1-\nu^2\right)^2}{\varphi^2} \frac{1}{\pi^2 \, D^2 \, n^2 \, \dot{m}} \int_{A_2} \frac{\rho}{2} \, c_{m2}(r)^2 \, d\dot{m} \tag{8}$$

$$\alpha_u = \frac{2 \, \left(1-\nu^2\right)}{\Psi_f^2 \, \ln \frac{1}{\nu}} \frac{1}{\pi^2 \, D^2 \, n^2 \, \dot{m}} \int_{A_2} \frac{\rho}{2} \, c_{u2}(r)^2 \, d\dot{m} \tag{9}$$

The ratio of η_{fs} to η_f is expressed with these definitions as

$$\frac{\eta_{fs}}{\eta_f} = 1 - \frac{1}{\left(1-\nu^2\right)^2} \frac{\varphi^2}{\Psi_f} \, \alpha_m - \frac{\ln \frac{1}{\nu}}{2 \, \left(1-\nu^2\right)} \, \Psi_f \, \alpha_u. \tag{10}$$

For a given design point φ, Ψ the hub-to-tip ratio ν and the profile parameters α_m, α_u have to be chosen to find the ratio of static to total efficiency. The profile parameters depend on the swirl distribution of the impeller. In scope of the study the common Free-Vortex Design is considered. In order to find the maximum efficiency depending on the design point, Strscheletzky's optimum criteria [1] for the hub-to-tip ratio ν is chosen.

The theoretical swirl distribution of Free-Vortex design is described by

$$c_u(r) \cdot r = const. \tag{11}$$

This distribution leads to a constant work at all radii and therefore to an uniform outflow c_m, if the inflow profile is isoenergetic and axially parallel. The flow is assumed to be inviscid. The c_m component can be computed as

$$c_m = c_{ax} = \frac{\dot{V}}{A_2} = \varphi \, \frac{\pi D n}{\left(1-\nu^2\right)}. \tag{12}$$

A detailed derivation of the profiles can be found in [5]. The profile coefficients are determined with these information. It is distinguished bewteen fans without and with outlet guide vanes OGV. OGVs lead to an elimination of the swirl ($\alpha_u = 0$). The resulting parameters can be found in table 1.

Table 1: Profile coefficients of Free-Vortex fans

Fan:	without OGV	with OGV
α_m	1	1
α_u	1	0

The hub-to-tip ratio ν is a very important parameter. The optimal ratio should lead to the lowest achievable outlet velocity c_2 and therefore minimal losses Ψ_{d2}. Furthermore it must be considered, that if the ratio is too small, the flow cannot follow the contour of the hub. The flow separates and this causes further losses.

In 1958 Strscheletzky [1] developed a criteria for optimal hub-to-tip ratio ν^*. He considered a swirling flow with a forced vortex core and a free vortex in the outer area. The optimal hub diameter has the dimension of the vortex core in a swirling flow. He distinguished between an axially unlimited and a limited swirling flow. This corresponds to fans without OGV and with OGV. The optimal hub-to-tip ratio for a given design point φ/Ψ is expressed by the following formulas

$$\text{limited swirl flow} \quad \left(\frac{\varphi}{\Psi_f}\right)_{opt} = \frac{1-\nu^{*2}}{2\nu^*} \tag{13}$$

$$\text{unlimited swirl flow} \quad \left(\frac{\varphi}{\Psi_f}\right)_{opt} = \frac{1}{2}\sqrt{\frac{1}{2}\left(\frac{1}{\nu^*}\right)^2 (1-\nu^{*2})^2 - (1-\nu^{*2}) \ln\left(\frac{1}{\nu^*}\right)}. \tag{14}$$

In figure 1 the curves for the optimal hub diameter ν^* are shown. The curve for the fans with OGV is higher than without OGV. This means that at a certain design point φ/Ψ a larger hub is necessary if OGV are installed.

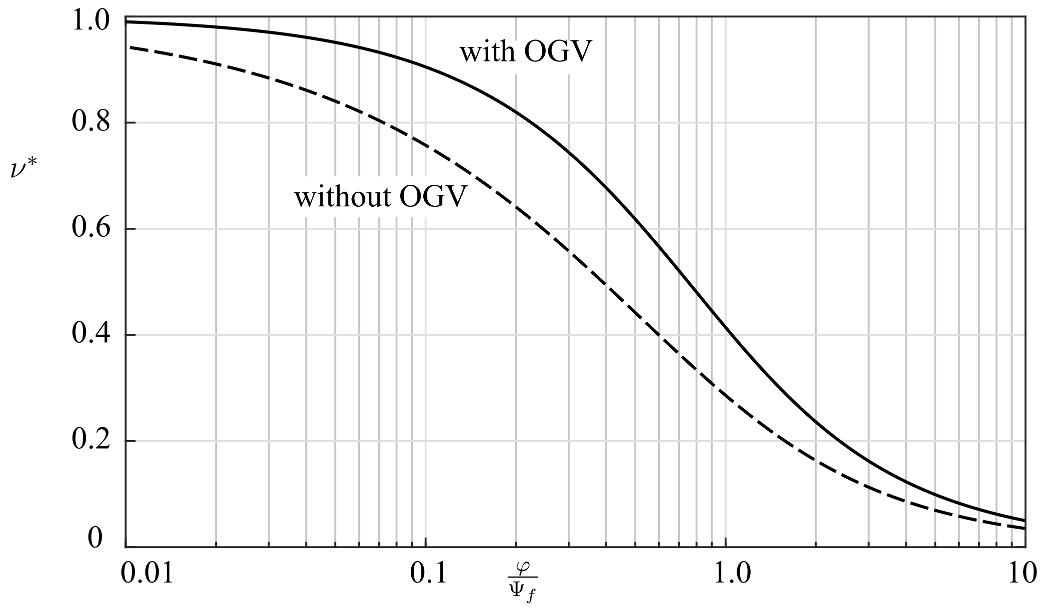

Figure 1: Opitmal hub-to-tip ratio $\nu*$ according to Strscheletzky

RESULTS

This section is divided into three subsections. First, the maximum achievable efficiency ratio η_{fs}/η_f is analyzed for fans without outlet guide vanes. In the second part fans with outlet guide vanes are discussed. The presentation and derivation of the results is based on [7]. In the last part, fans with outlet guide vanes and diffusers are investigated.

Fan without outlet guide vanes

The efficiency ratio η_{fs}/η_f is reduced by the losses due to the c_m and c_u components of the velocity. Figure 2 shows contour lines of the efficiency η_{fs}/η_f and the hub-to-tip ratio ν^* (dashed line) for the design points φ, Ψ_{fs}. The results are based on equation 10 with the profile coefficients in table 1.

As expected, the achievable efficiency decreases for higher volume numbers. The φ, Ψ_{fs} combinations are limited. The design points above the bold dashed line cannot be realized. An increase of the pressure number ψ_{fs} at a constant volume number φ leads to a higher recommended hub-to-tip ratio ν^*. This can be lead back to increasing forced vortex core in Strscheletzkys model due to the higher blade load.

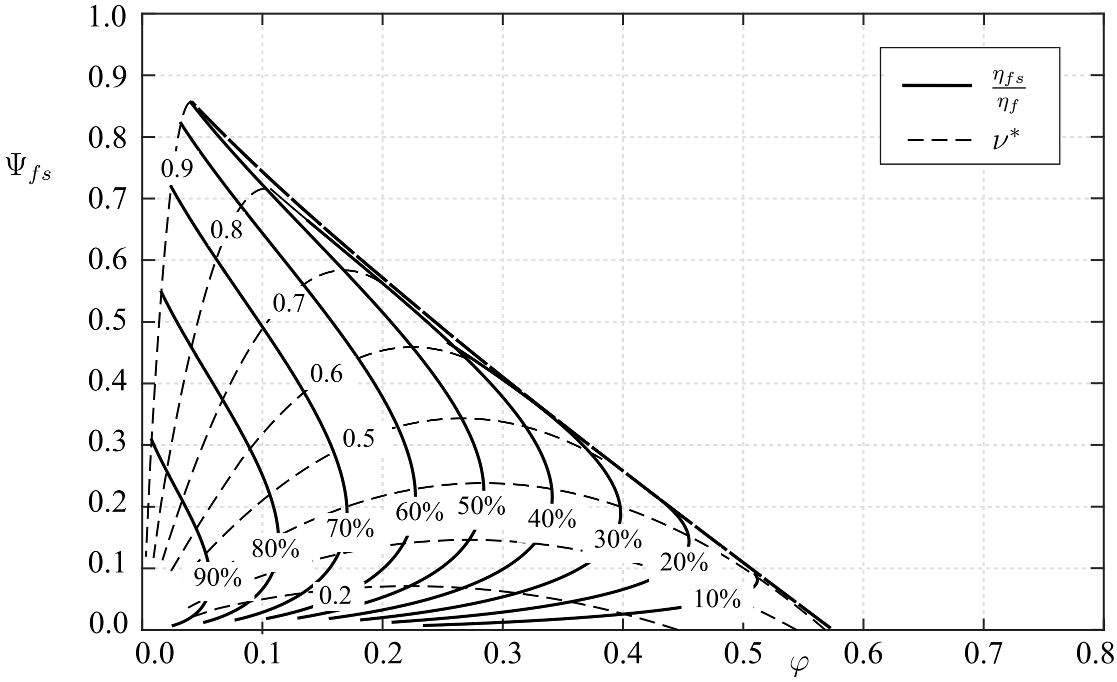

Figure 2: Fan without outlet guide vanes

Fan with outlet guide vanes

If outlet guide vanes are installed, the swirl is converted into a further pressure rise. This leads, compared to the fan without OGV, to a higher efficiency ratio η_{fs}/η_f at identical design points.

In figure 3 the achievable efficiency and recommended hub-to-tip ratio are shown.

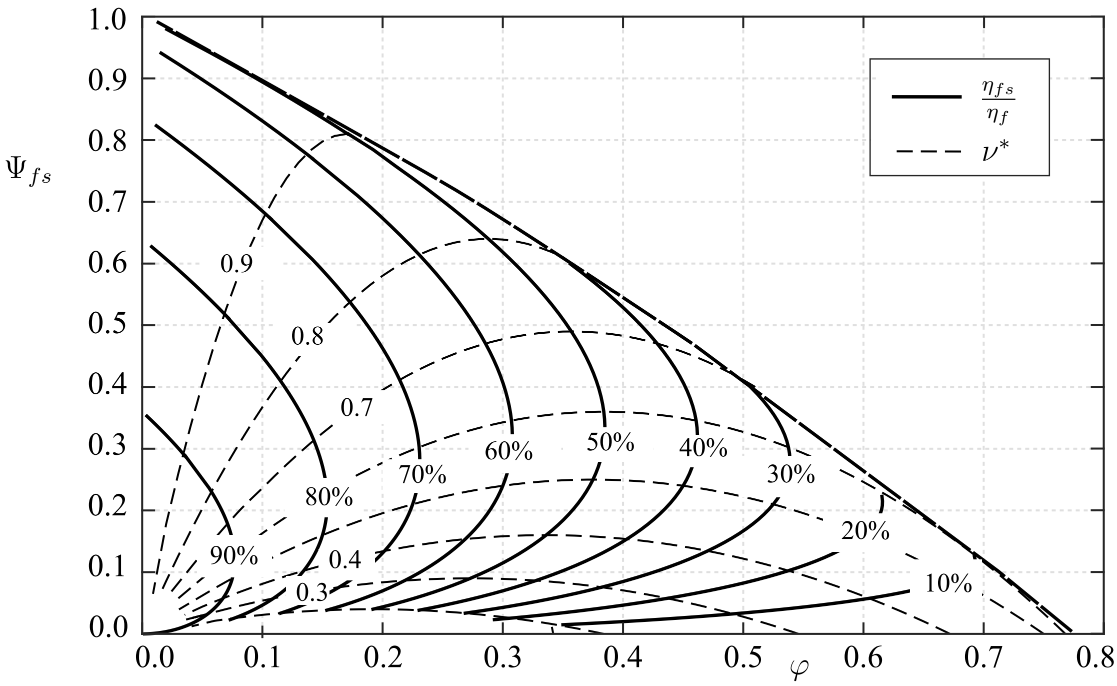

Figure 3: Fan with outlet guide vanes OGV

Fans with OGV can be designed for operation points with higher pressure numbers. The bold dashed line, which indicates the limit, is located above the corresponding line in figure 2. For the common design point $\varphi = 0.3$, $\Psi_{fs} = 0,25$ the maximum achievable efficiency is $\eta_{fs}/\eta_f = 60\%$. This is an increase of 7 percentage points compared to the fan without OGV.

Fan with outlet guide vanes and diffuser

A sketch of the fan with diffuser is displayed in figure 4. The rotor RO is followed by the outlet guide vanes OGV and the annular diffuser DF.

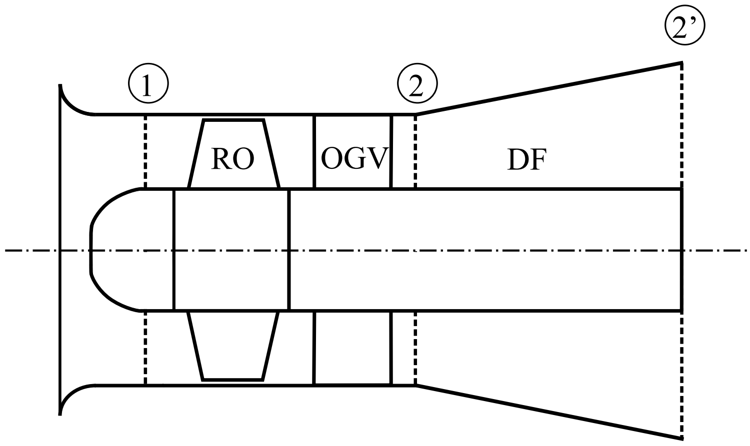

Figure 4: Fan with outlet guide vanes OGV and diffuser DF

The diffuser DF is a separate component which does not belong directly to the fan. However, the efficiency η_f respectively the pressure height Ψ_f are generally related to the fan outlet (2) by the manufacturer. Therefore the efficiency ratio is defined with $\eta_{fs'}/\eta_f$ (eq.16). The value $\eta_{fs'}$ includes the fan and the diffuser DF (1 - 2'), η_f only the fan (1 - 2). The outlet losses are reduced by the pressure recovery in the diffuser, which is described by the pressure recovery coefficient cp (eq.15).

$$cp = \frac{p_{2'} - p_2}{p_{d2}} = \frac{\Psi_{2'} - \Psi_2}{\varphi^2/(1-\nu^{*2})^2} \tag{15}$$

$$\frac{\eta_{fs'}}{\eta_f} = 1 - \frac{1}{(1-\nu^2)^2}\frac{\varphi^2}{\Psi_f}(1-cp) \tag{16}$$

The pressure recovery cp of diffusers was studied by Sovran and Klomp [8]. They investigated the pressure recovery for various diffuser geometries and examined the influence of the inflow boundary layer thickness. The thickness is described by the blocked area fraction $B = 1 - 1/A \int_A c/c_{max}\, dA$. They showed, that the optimum lines cp^* are nearly identical over the range of thicknesses $B = 0.02 - 0.18$. They developed diffuser design charts, where the optimum pressure recovery cp^* for a given diffuser length L_{DF} can be determined. This chart for annular diffuser is shown in figure 5. The contour lines give the pressure recovery cp for a non-dimensional diffuser length $L_{DF}/\Delta r_2$ and area ratio $A_{2'}/A_2$.

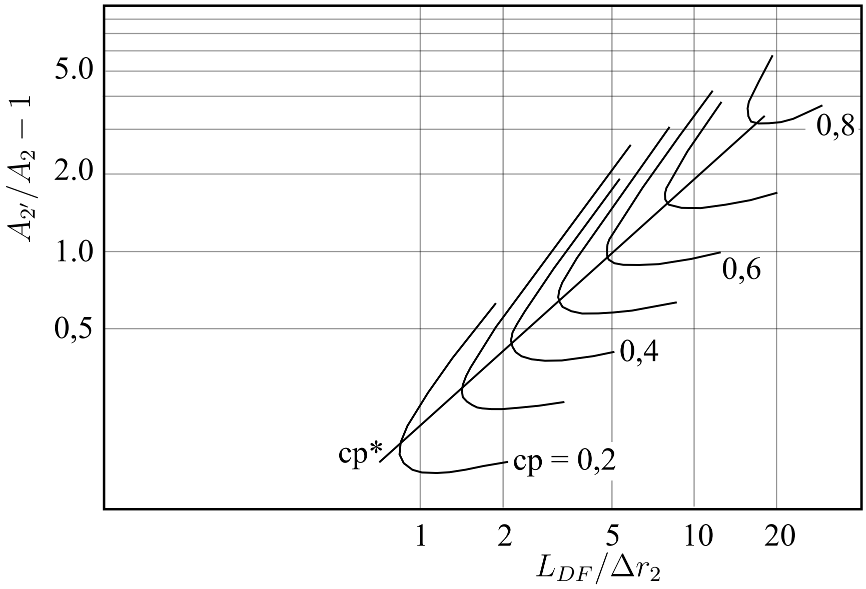

Figure 5: Annular diffuser performance chart [8]

The optimum line cp^* is approximated by the function

$$cp = a - b\, e^{-c\, L_{DF}/\Delta r_2}. \tag{17}$$

The coefficients are $a = 0.7954$, $b = 0.7497$ and $c = 0.2921$. In order to evaluate the quality of the fit, the coefficient of determination R^2 is evaluated. The function shows an agreement with $R^2 = 0.9978$. The maximum achievable efficiency $\eta_{fs'}/\eta_f$ is shown in figure 6. The installation of the diffuser DF leads to a significant increase of the pressure number $\Psi_{fs'}$. All design points in the range from $\varphi = 0...1$ and $\Psi_{fs'} = 0...1$ can be realized. At the design point $\varphi = 0.3$, $\psi_{fs} = 0.25$ the maximum achievable efficiency η_{fs}/η_f is 80%. This is an increase of 20 repectively 27 percentage points compared to the latter configurations.

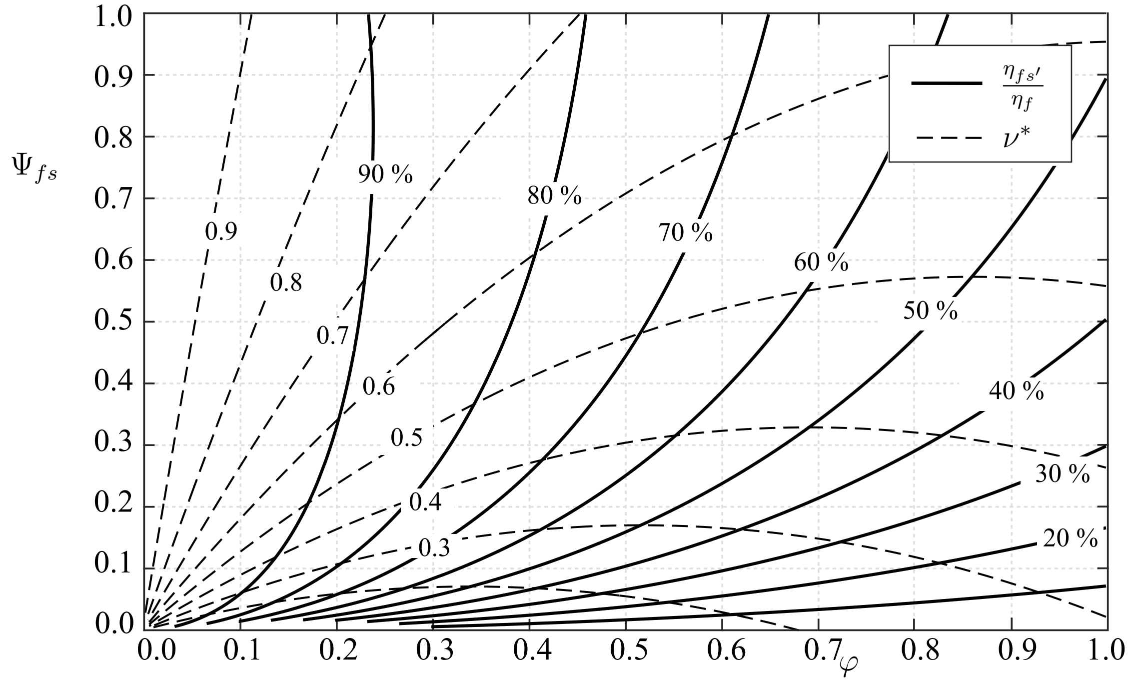

Figure 6: Fan with outlet guide vanes OGV and diffuser DF

CONCLUSIONS

The total to static efficiency η_{fs}/η_f of a fan can be significantly raised with OGV or a diffuser. However it should be taken into account, that additional components (OGV, DF) increase the length and therefore the costs of the product. The diffuser length in the study corresponds to the fan diameter. At the common design point $\varphi = 0.3$, $\Psi_{fs} = 0.25$ the installation of OGV leads to an efficiency increase of 7 % percentage points. If the diffuser is added additionally, the increase accounts for 27 percentage points. It is shown, that the number of realizable design points (φ, Ψ) depends on the design of the fan. The installation of OGV and/or a diffuser leads a larger number of realizable φ, Ψ combinations.

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