1. INTRODUCTION

1.1. Classification, restriction of topic under discussion

Fluid:

- **Gas**
- (Liquid)
- (Multiphase fluid)

Power input / output:

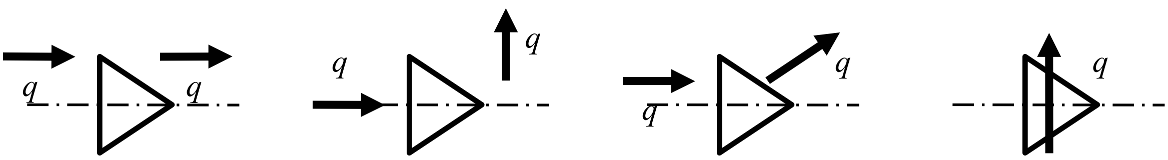
- Power input – transportation of fluid from a domain of lower pressure („suction side”) to a domain of higher pressure („pressure side”)
- (Power output – e.g. wind turbines)

Operating principle:

- **Euler principle (fluid mechanical principle): TURBOMACHINERY**. Bladed rotor in a casing. Mechanically free passing between suction and pressure sides.
- (Volumetric principle: operating enclosure between the suction and pressure sides, bounded by stationary and moving walls. No mechanically free passing. E. g. piston compressors.)

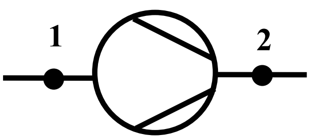
1.2. Classification of fluid machinery for power input

1.2.1. Direction of through-flow relative to the axis of rotation:



axial (radial) (mixed) (cross-flow)

1.2.2. Pressure increase, pressure ratio:



p_2/p_1 **pressure ratio**

A/ $p_2/p_1 < 1.1$ (1.2) fans

$\rho \approx$ constant, $\Delta T \approx 0$ design, constructional, application aspects

JUSTIFICATION: Isentropic approach:

$$p_2/p_1 = (\rho_2/\rho_1)^{\chi} \qquad (1.1)$$

Substituting p_2/p_1 =1.1 , for air, the result is ρ_2/ρ_1 =1.07 , being still negligible according to the above aspects.

Being constant of density gives criteria for the fluid velocity characterising the machinery, e.g. for the circumferential velocity of the blade tip, or for the fluid velocity developing int he connected machinery. Let us consider a streamline between the S point of the suction port and the stagnation point T located on the nose cone of the machine.

Having criteria being even more strict compared to the previous relative difference of density of 7 %:

$$\left|\rho_T - \rho_S\right| \le 0.05\rho_S \qquad (1.2)$$

The energy equation between points S and T:

$$T_S + \frac{v_S^2}{2c_p} = T_S\left(1 + \frac{\chi R {Ma_S}^2}{2c_p}\right) = T_T \qquad (1.3)$$

Where the following relationship was considered:

$$Ma_S = \frac{v_S}{a_S} = \frac{v_S}{\sqrt{\kappa R T_S}} \qquad (1.4)$$

Isentropic change of state:

$$\frac{\rho_T}{\rho_S} = \left(\frac{T_T}{T_S}\right)^{\frac{1}{\kappa-1}} \qquad (1.5)$$

With assumption of $\left|\rho_T - \rho_S\right| = 0.05\rho_S$ and with substitution of air characteristics, the combination of Eqs. (1.3) and (1.5) reads [1]:

$$Ma_S < 0.31 \qquad (1.6)$$

This condition reads, with assumption of air of room temperature, that no velocities higher than 100 *m/s* are allowed to develop in the machinery and in the connected system. The circumferential velocity of the blade tip characterises of the machine in a lifelike manner.

Summary: **FOR FANS IN GENERAL**

- **Assuming nearly atmospheric pressure on the suction side, Δp < 0.1 *bar*** [2]
- **Blade tip circumferential speed < 100 *m/s*, being in harmony with the mechanical and acoustic demands related to the turbomachinery.**

B/ 1.1 < p_2/p_1 < 3 blowers

$\rho \neq$ constant, $\Delta T > 0$, but cooling due to natural convection is still sufficient.

C/ 3 < p_2/p_1 compressors

$\rho \neq$ constant, $\Delta T >> 0$, artificial cooling is necessary. (Mechanical and technological aspects)

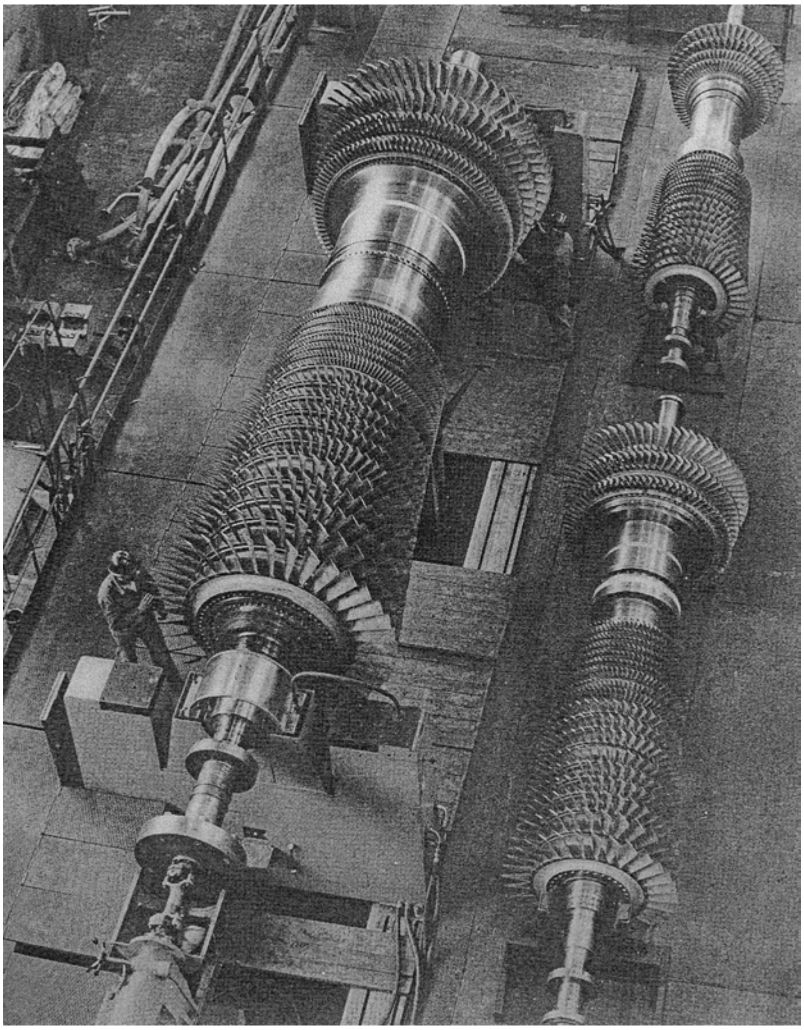


Fig. 1.1. Industrial gas turbine [4]

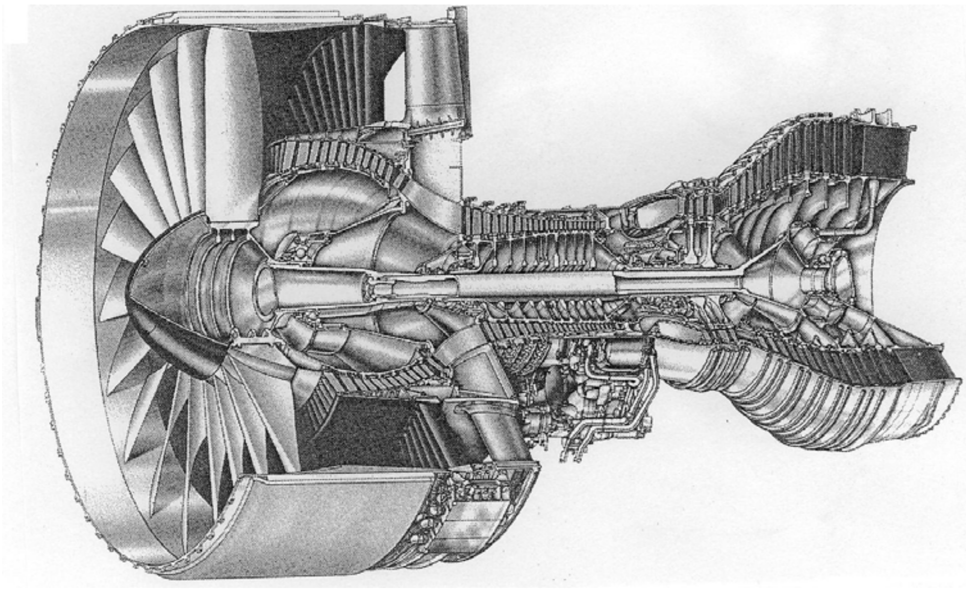
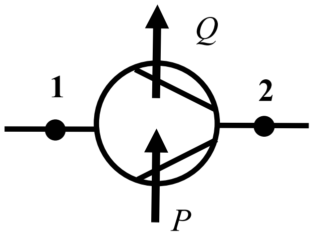


Fig. 1.2. Airplane jet combustion engine [4]

1.3. Work process of fans



For ideal (inviscid) case:

$$\frac{P}{q_m} - \frac{Q}{q_m} = \left[\frac{v^2}{2} + gh + U + \frac{p}{\rho}\right]_1^2 = i_{t2} - i_{t1} \tag{1.7}$$

For fans $Q = 0$, $\Delta U = 0$, and gh plays role only if the densities of the ambient and the transported fluid differ (e.g. transportation of hot fume gas or cold air).

Therefore, for fans in general

$$P = q_m\left[\frac{v^2}{2} + \frac{p}{\rho}\right]_1^2 = q_V\left[\rho\frac{v^2}{2} + p\right]_1^2 = q_V \Delta p_{tid} \tag{1.8}$$

Where, due to the Euler equation of turbomachines [3]:

$$\Delta p_{tid} = \rho\left(\underline{v}_2\underline{u}_2 - \underline{v}_1\underline{u}_1\right) \tag{1.9}$$

Departures from ideal (inviscid) case:

A/ Due to volumetric losses, the volume flow rate q_{VR} through the rotor is higher than the volume flow rate q_V utilised (a percentage circulates inside the rotor), but extra power is to be introduced for circulation of the extra amount. This effect is considered with use of the volumetric efficiency η_V:

$$P = q_{VR}\Delta p_{tid}\ ,\ q_{VR} > q_V\ ,\ \eta_V = q_V / q_{VR} \tag{1.10}$$

B/ The total pressure rise is less than the ideal value, due to friction losses. This effect is considered with use of the hydraulic efficiency η_h, also called as **total efficiency** (because of its relation to total pressure rise):

$$\eta_h = \Delta p_t / \Delta p_{tid} \tag{1.11}$$

With use of the above

$$P = q_{VR}\Delta p_{tid} = \frac{q_V}{\eta_V}\frac{\Delta p_t}{\eta_h} = \frac{P_{useful}}{\eta_V \eta_h} \tag{1.12}$$

C/ The overall input power covers the mechanical losses, e.g. losses of belt drive and bearing, as well. This effect is considered with introduction of the mechanical efficiency:

$$\eta_m = P / P_{overall} \tag{1.13}$$

Therefore $P_{overall} = \dfrac{P_{useful}}{\eta_V \eta_h \eta_m} = \dfrac{P_{useful}}{\eta_{overall}}$ (1.14)

For fans, it is usually true that $\eta_m \approx 1$, $\eta_V \approx 1$ (no considerable pressure difference occurs; this would not be true in the case of pumps), and therefore, $\eta_{overall} \approx \eta_h$.

1.4. Basic construction of axial flow fans

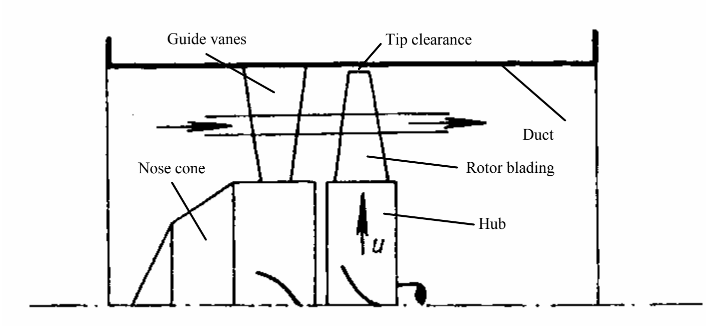


Fig. 1.3. Sketch of an axial fan [2]

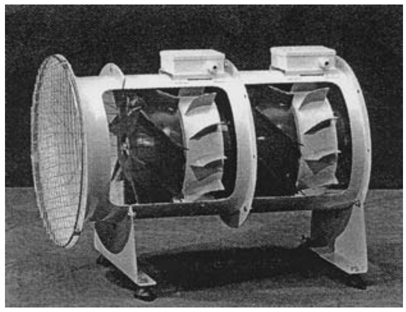


Fig. 1.4. Axial (mixed-flow ?) fan [6]

Characteristic geometrical data: inner (hub) and outer (tip) diameters of rotor blading: D_1 , D_2 , respectively.

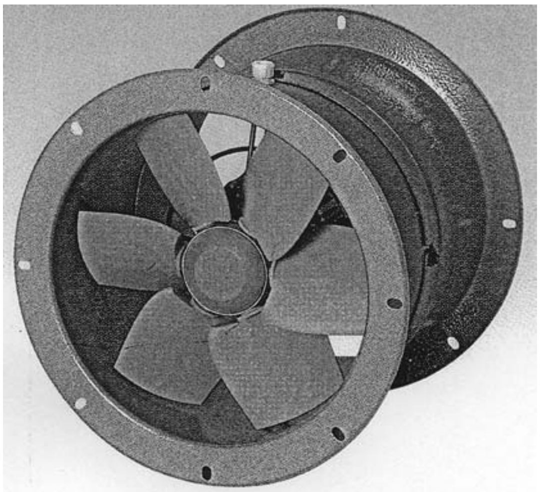


Fig. 1.5. Ducted axial fan [5]

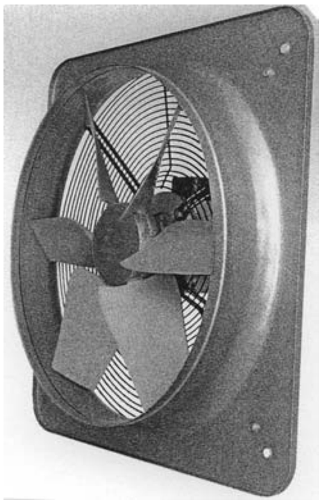


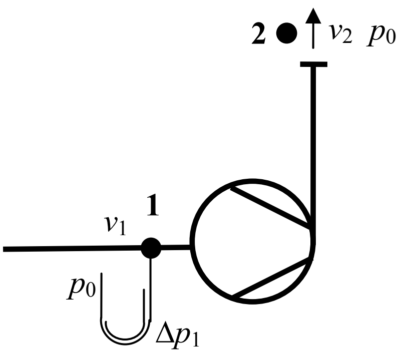
Fig. 1.6. Wall axial fan [5]

PRINCIPAL DIFFERENCES BETWEEN RADIAL AND AXIAL TURBOMACHINES:

- According to Euler equation of turbomachines (1.9), in the case of axial flow machinery $r_1 \approx r_2$, $u_1 \approx u_2$. For this reason, the axial flow machines usually achieve less total pressure rise than radial flow machines.
- In the case of fixed fluid mechanical performance, this also means that axial turbomachines usually perform higher volume flow rate than radial flow turbomachines.
- In an axial flow turbomachine, the fluid flow deflection is reduced compared to that of a radial flow machine. This leads to reduced losses and increased efficiency of the axial flow turbomachine unit.
- In the case of an axial flow turbomachine, the through-flow direction accommodates the flow direction in the connected duct system. This leads to reduced losses not only in the turbomachine in itself but also in the connected system.

1.5. Axial fan arrangements: dependent on technology and geometry!

1.5.1. From duct to the surroundings (extraction, "induced flow", e.g. food industry: vapour and stink extraction)



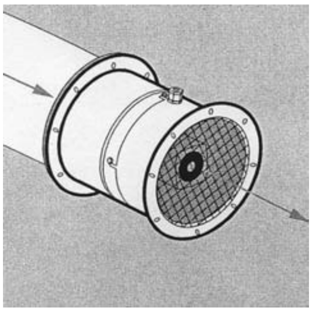


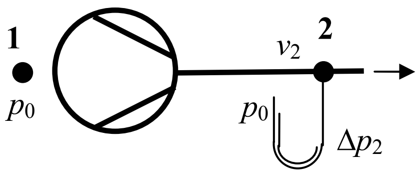
Fig. 1.7. "From duct to the surroundings" arrangement [5]

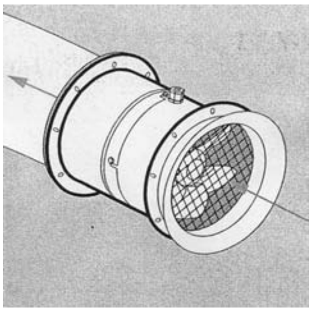
$$\Delta p_t = \left(\rho\frac{v_2^2}{2} + p_0\right) - \left[\rho\frac{v_1^2}{2} + (p_0 - \Delta p_1)\right] = \Delta p_1 + \rho\frac{v_2^2}{2} - \rho\frac{v_1^2}{2} \tag{1.15}$$

(=Δp_1 for axial fans, because $v_2 = v_1$ due to the uniformity of the fluid cross-section)

$$\Delta p_{st} = \Delta p_t - \rho\frac{v_2^2}{2} = \Delta p_1 - \rho\frac{v_1^2}{2} \tag{1.16}$$

1.5.2. From surroundings to the duct (inflow, "forced flow", e.g. boiler feed air fan, or fans providing overpressure: clean room technology)





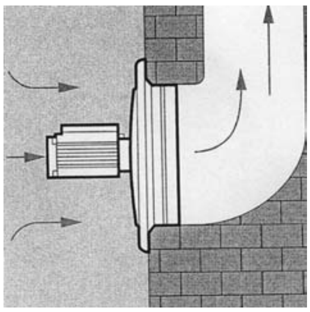


Fig. 1.8. "From surroundings to the duct" arrangement [5]

With neglect of losses on the suction side:

$$\Delta p_t = \left[\rho\frac{v_2^2}{2} + (p_0 + \Delta p_2)\right] - p_0 = \Delta p_2 + \rho\frac{v_2^2}{2} \tag{1.17}$$

$$\Delta p_{st} = \Delta p_2 \tag{1.18}$$

1.5.3. From duct to duct (ducted fan: upstream and downstream of it: service elements. E.g. upstream: calorifer, downstream: jeallousie)

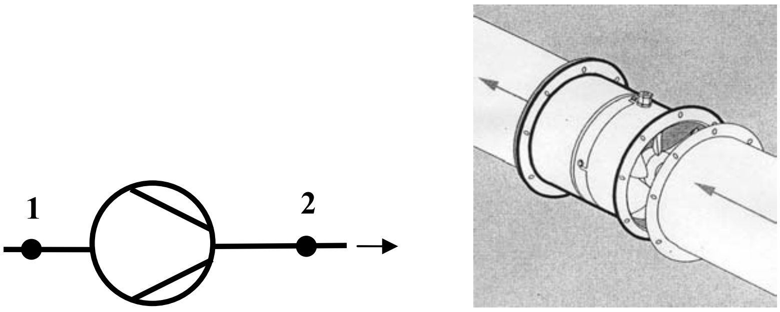


Fig. 1.9. "From-duct-to-duct" arrangement [5]

$$\Delta p_t = \left(\rho \frac{v_2^2}{2} + p_2\right) - \left(\rho \frac{v_1^2}{2} + p_1\right) \tag{1.19}$$

$$\Delta p_{st} = p_2 - \left(\rho \frac{v_1^2}{2} + p_1\right) \tag{1.20}$$

1.5.4. From surroundings to surrounding (e.g. tunnel ventilation)

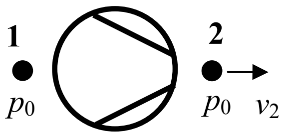
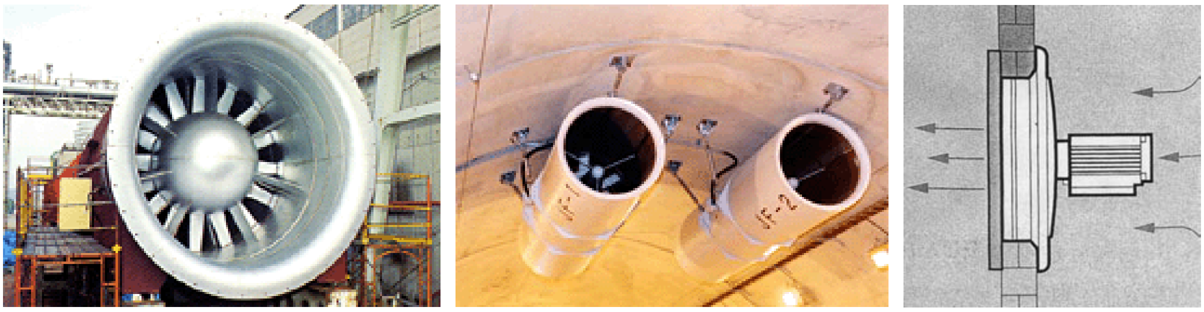


Fig. 1.10. [5]



$$\Delta p_t = \left(\rho \frac{v_2^2}{2} + p_0\right) - p_0 = \rho \frac{v_2^2}{2} \tag{1.21}$$

$$\Delta p_{st} = \Delta p_t - \rho \frac{v_2^2}{2} = 0 \tag{1.22}$$

Even in absence of duct, the static pressures may be different on the two sides!

1.6. Characteristic curve: an example

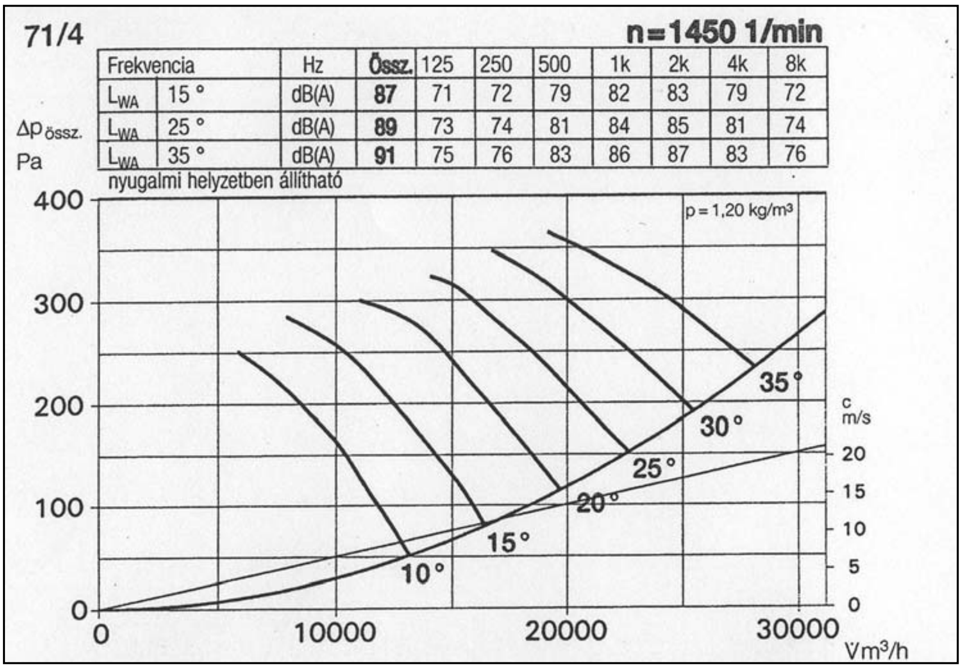


Fig. 1.11. Characteristic curves of a HELIOS HQ 71/4 axial fan [6]

1.7. Dimensionless characteristics, comparison

- User demands: Δp_t, (Δp_{st}), q_V, $P_{overall}$ (motor selection)
- Machinery characteristics: D (rotor outer diameter), n
- Fluid characteristics: ρ, ν

To compare various machines: ($u_t = D\,\pi\,n$)

Total pressure coefficient: $$\Psi_t = \frac{\Delta p_t}{\frac{\rho}{2}u_t^2} \tag{1.23}$$

Static pressure coefficient: $$\Psi_{st} = \frac{\Delta p_{st}}{\frac{\rho}{2}u_t^2} \tag{1.24}$$

Flow coefficient: $$\Phi = \frac{q_v}{A_{char}\;u_t} \tag{1.25}$$

Where $A_{char} = (D_{tip}^2 - D_{hub}^2)\pi/4 = = D_{tip}^2\,\pi/4\,(1-\nu^2)$ annulus cross-section

Hub-to-tip ratio: $\nu = D_{hub} / D_{tip}$ (1.26)

Power coefficient: $$\lambda = \frac{P_{overall}}{\frac{\rho}{2}u_t^2 A_{char} u_t} = \frac{\frac{\Delta p_t q_V}{\eta_t}}{\frac{\rho}{2}u_t^2 A_{char} u_t} = \frac{\Psi_t \Phi}{\eta_t} \tag{1.27}$$

Reynolds number for axial fans: $$\mathrm{Re} = \frac{u_t \ell}{\nu} \quad (1.28)$$

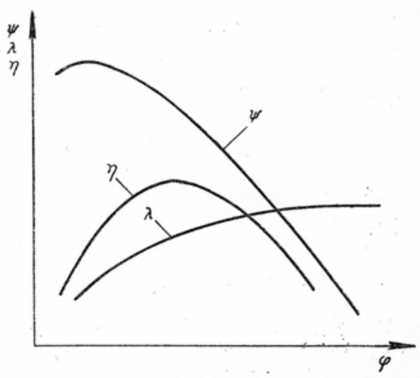


Fig. 1.12. Dimensionless characteristics [2]

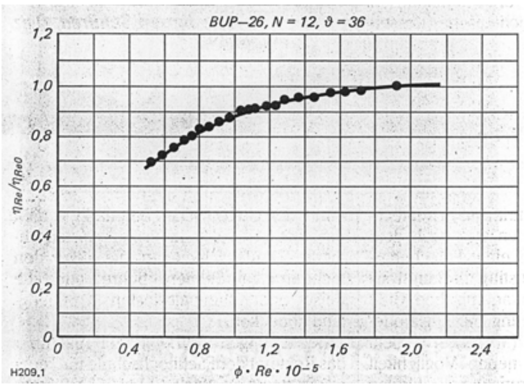


Fig. 1.13. Reynolds number dependency of hydraulic efficiency of an axial fan [7]

For axial fans: the (1.28) Reynolds number is to be kept above 300.000 (critical Reynolds number) (… above 100.000)

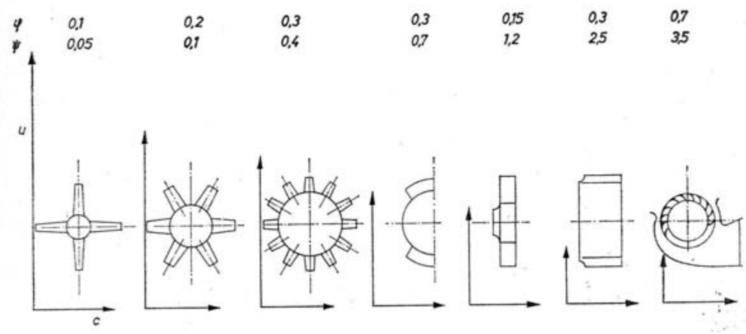


Fig. 1.14. Comparison of various fans [2]

LIST OF SYMBOLS

a	$[m/s]$	sound speed
c_p	$[J/(kgK)]$	isobar specific heat (= 1005 $J/(kgK)$ for air)
D	$[m]$	diameter
g	$[N/kg]$	gravity field intensity
h	$[m]$	level, height
i_{t}	$[J/kg]$	total entalphy
ℓ	$[m]$	chord length
Ma	[-]	Mach number
n	$[1/s]$	rotor speed
q_{m}	$[kg/s]$	mass flow rate
P	$[W]$	input shaft power
P_{useful}	$[W]$	useful fluid mechanical power
$P_{overall}$	$[W]$	total input power
p	$[Pa]$	static pressure
p_{t}	$[Pa]$	total pressure
Δp_{t}	$[Pa]$	total pressure rise
Δp_{st}	$[Pa]$	static pressure rise
Q	$[W]$	heat power extracted by cooling
q_{V}	$[m^3/s]$	volume flow rate
R	$[J/(kgK)]$	specific gas constant (= 287 $J/(kgK)$ for air)
r	$[m]$	radius
T	$[K]$	temperature
U	$[J/kg]$	specific internal energy
u	$[m/s]$	circumferential speed
u_{t}	$[m/s]$	rotor blade tip circumferential speed
v	$[m/s]$	absolute velocity
Δ		difference between characteristics of pressure and suction sides
ν	$[m^2/s]$	kinematic viscosity
ρ	$[kg/m^3]$	density
χ	[-]	isentropic exponent (= 1.40 for air)
η_{V}	[-]	volumetric efficiency
η_{h}	[-]	hydraulic efficiency
η_{m}	[-]	mechanical efficiency
η_{overall}	[-]	overall efficiency

SUBSCRIPTS:

0	characteristics of atmosphere
1	suction side; fluid entering the rotor
2	pressure side; fluid exiting the rotor
id	ideal (no losses)