LEGENDA / *LEGEND*

Ps	*Static pressure (mm/H_2O - Pa)*
Pd	*Dynamic pressure (mm/H_2O - Pa)*
Pt	*Total pressure (mm/H_2O - Pa)*
Q	*Air delivery (m^3/h)*
U	*Rated voltage (V)*
M	*Rated voltage and frequency single-phase (230V-50Hz)*
T	*Rated voltage and frequency three-phase (400V-50Hz)*
rpm	*Nominal motor speed*
Pm	*Installed motor power (kW)*
In	*Maximal absorbed current (A)*
IP	*Motor mechanical protection*
Cl	*Motor Insulation class*
S	*Outlet area (m2)*
C	*Air velocity (m/s)*
Pd2	*Impeller inertia moment (Kgm2)*
Lp	*Sound pressure level (dB)*
Lw	*Sound power level (dB)*
Reg.	*Speed regulator*
P	*n° Poles*
2 poles	*3000 nominal rpm*
4 poles	*1500 nominal rpm*
6 poles	*1000 nominal rpm*
8 poles	*750 nominal rpm*

Attention: the sound pressure level is measured in free field, with inlet and outlet sides connected to duct.

Note: when selecting it is necessary to keep in mind that the noise of the fan, if not measured in free field, will be higher than the shown values.

Standards achieved: Performance tests according to EN ISO 5801 standard - Acoustic tests according to EN60651 standard.

This section is intended to give a brief overview of the most common technical aspects of industrial, commercial and domestic ventilation. For more details please refer to specialized publications.

1. DEFINITIONS

Fan performance are usually expressed through a characteristic curve that, for a certain revolution per minute of the impellor and for specific air conditions, provides the value of the static pressure (ps), or total pressure (pt), given the volumetric flow rate (Q).
Also the value of the absorbed mechanical power (Pw) and the efficiency (n) are expressed given the volumetric flow rate (Q).

1.1 FLOW RATE

Volumetric flow rate is the fluid volume that pass through the fan in a specific amount of time. It is normally expressed in cubic meters per hour (m3/h), cubic meters per second (m3/s) or cubic feet per minute (cfm).
The relationship between the flow rate of a ducted fan, the speed of the fluid in the duct and the cross section area of the duct itself is expressed by the following formula:

$$V = \frac{Q}{A \cdot 3600} \qquad (1.1)$$

where:
V = fluid average speed in m/s
Q = flow rate in m^3/h
A = duct cross section area in m^2

1.2 STATIC, DYNAMIC AND TOTAL PRESSURE

When a fluid is in movement, there are three types of pressure involved. Pressure is normally expressed in pascal (Pa), millimetres of water gate (mmH2O o mmWG) or inches of water gate (inWG).

Static pressure (ps)
It is defined as the pressure applied by the fluid to the wall of the duct or of the container in which it's hold. It expresses the potential energy suitable to overcome the resistance given by the system to the fluid transit.
It acts equally in every direction and it is independent from fluid speed.
With reference to environment pressure, static pressure is positive when it is bigger than ambient pressure and negative when it is lower.

Dynamic pressure (pd)
It is defined as the pressure related to the energy that belongs to the mass unit of the fluid because of its speed. It expresses the kinetic energy of the fluid in movement. It acts in the same direction of the fluid movement and it is always positive.
The dynamic pressure is function of fluid speed and density and is expressed by the following formula:

$$pd = 1/2 * \rho * V^2 \qquad (1.2)$$

where:
pd = dynamic pressure in Pa
ρ = fluid density in kg/m^3
V = fluid speed in m/s

Total pressure (pt)
It is defined as the algebraic sum of static pressure (ps) and dynamic pressure (pd):

$$pt = ps + pd \qquad (1.3)$$

When a fan is operating with shut inlet or outlet, flow rate is null. So fluid speed and, consequently, dynamic pressure are null. Therefore the result is:

$$pt = ps$$

This working condition corresponds to the beginning point (on the left) of the fan performance curve.

When a fan is operating with free inlet and outlet (both inlet and outlet not ducted), static pressure is null. Therefore the result is:

$$pt = pd$$

This working condition corresponds to the ending point (on the right) of the fan performance curve.

1.3 ABSORBED POWER AND EFFICIENCY

A fan needs some mechanical power to give a certain flow rate at a certain total pressure.
This mechanical power depends also from the aeraulic efficiency of the fan and is given by the following formula:

$$Pw = \frac{Q * pt * 100}{\eta} \qquad (1.4)$$

where:
Pw = mechanical power in W
Q = flow rate in m3/s
pt = total pressure in Pa
η = fan aeraulic efficiency in %

Mechanical power is given by an electric motor, that also absorb a certain electrical power from power supply network. Following formulas are commonly used:

- three-phase motor

$$Pe = V * I * \sqrt{3} * \cos\varphi = \frac{Pw}{\eta_{mot}} \qquad (1.5)$$

- mono-phase motor

$$Pe = V * I * \cos\varphi = \frac{Pw}{\eta_{mot}} \qquad (1.6)$$

where:
Pe = electrical power absorbed from power supply network in W
Pw = mechanical power in W
I = absorbed power in A
V = supply voltage in V
η_{mot} = motor efficiency in %

1.4 WORKING POINT

The energy that a fan receives from the electric motor is transferred to the volume unit of fluid in transit in the form of pressure. The pressure that a fan can give is not constant, though, but is function of the flow rate according to the fan characteristic curve. Also the absorbed power varies with the flow rate.

To have a certain amount of air circulating in a system, it is necessary to apply to the fluid a certain energy, in the form of pressure, to win the frictions encountered in the movement. The pressure to be applied varies with the flow rate and is given by the following formula:

$$p = Kr * Q^2 \qquad (1.7)$$

where:
p = pressure needed by the system
Kr = factor depending on system characteristics
Q = airflow

Kr factor remains constant for variations not too big of the airflow and can be obtained starting from the 1.7 in a certain working point, calculated or measured.
Once Kr factor is obtained, it is possible to draw a curve of p against Q, that is the system characteristic curve.

A fan installed in a system will give the airflow corresponding to the value of static pressure necessary to overcome the resistance to the air movement in the system.
If on the same diagram are drawn both the static pressure curve of the fan (fan characteristic curve) and the aeraulic resistance curve (system characteristic curve), the crossing point of the two curves will be the working point (see Fig 1).

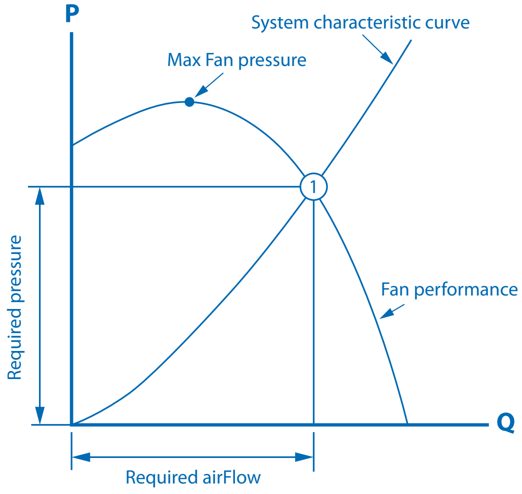


FIG. 1

1.5 VARIATION IN AIR DENSITY

Fans characteristics in Dynair catalogue refers to working conditions with air at 15°C and at 0 m above sea level, that means with density ρ = 1,225 kg/m3.
In case working conditions are different from standard, it is recommended to adjust required performance in order to make the selection with performance in standard conditions (+15 °C at 0 m above sea level).

To adjust the pressure (static or total) from real working conditions to standard conditions, the following formula is used:

$$p_0 = p_1 * \frac{\rho_0}{\rho_1} \qquad (1.8)$$

where:

p_0 = *pressure adjusted to standard density*

p_1 = *pressure required in real working conditions*

ρ_1 = *air density in real working conditions*

ρ_0 = *standard air density (1,225 kg/m3)*

The same relationship is valid for the fan absorbed power. Airflow, instead, remains substantially unchanged.

In section 6 of Dynair Industrial Catalogue there is a table with air density values against variation of temperature and altitude.

2 FANS CLASSIFICATION

Fans are rotation machines able to create a continuous air flow with an aerodynamic action.
There are substantially two types of fans, axials and centrifugals.
In axial fans the air stream is parallel to the impellor rotation axis and air discharge is in the same direction of air inlet. In centrifugal fans the air stream is pushed in radial direction from the impellor rotation axis and typically air discharge is at 90° from air inlet.

Axial fans can have the impellor with metal sheet blades (generally cut and pressed in shape) or with airfoil blades (generally in technopolimer or in aluminium alloy). There are fans for wall, duct or roof application. Generally, given static pressure is not high.

Centrifugal fans can have forward curved, backward curved or radial blades. There are several types: with volute casing, in-line, roof-fans…
Compared to axials, they normally give higher static pressure.

Forward curved centrifugal fans are normally small/medium items. They are cheap and with an efficiency not very high. As working point moves towards free inlet/outlet condition absorbed power grows rapidly, becoming even higher than the maximum value of the motor.
Double inlet versions, direct drive or belt driven, are normally used in box fans or air handling units.

Backward curved centrifugal fans have higher efficiency and can reach bigger dimensions. Absorbed power remains within motor limit, even in free inlet/outlet condition.

3 FAN LAWS

Performance of Dynair geometrically similar fans can be calculated using the following relationships between rotational speed, impeller diameter and air density.

3.1 Given a certain impellor diameter and air density and changing the rotational speed (rpm):

$$Q_2 = Q_1 * \left[\frac{rpm_2}{rpm_1}\right] \quad p_2 = p_1 * \left[\frac{rpm_2}{rpm_1}\right]^2 \quad Pw_2 = Pw_1 * \left[\frac{rpm_2}{rpm_1}\right]^3 \quad (3.1)$$

3.2 Given a certain rotational speed and air density and changing the impellor diameter (D):

$$Q_2 = Q_1 * \left[\frac{D_2}{D_1}\right]^3 \quad p_2 = p_1 * \left[\frac{D_2}{D_1}\right]^2 \quad Pw_2 = Pw_1 * \left[\frac{D_2}{D_1}\right]^5 \quad (3.2)$$

4 NOISE AND VIBRATIONS

Fans are rotating mechanical machines, so they inevitably generate noise and vibrations. The problem for designers or users, then, is not if the fan is noisy or vibrating, but how much it is noisy and vibrating, or rather if the noise and vibrations level is compatible with project needs.

4.1 SOUND POWER AND PRESSURE

The generation of a sound (or noise) is due to an item vibrations and the sound is propagated in every mean that can vibrate.
The sound source produces in the air little alternate pressure fluctuations around the barometric equilibrium pressure, causing compressions and de-compressions that are propagated creating a sound wave. The value (effective value) of such fluctuation is called sound pressure and is measured in pascal (Pa).
Human ear and any microphone sense the sound pressure itself.
Conventionally, it's been introduced the sound pressure level (Lp):

$$Lp = 20 * \log \frac{p}{p_0} \qquad (4.1)$$

where:
Lp = sound pressure level in dB
p = actual sound pressure in Pa
*p_o = reference sound pressure (2*10^-5 Pa)*

The sound emission by a certain machine implies a loss of a certain amount of energy. Such energy, referred to time measure unit, is the sound power and is measured in watt (W). Conventionally, it's been introduced the sound power level (Lw):

$$Lw = 10 * \log \frac{Pw}{Pw_0} \qquad (4.2)$$

where:
Lw = sound power level in dB
Pw = actual sound power in W
Pwo = reference sound power ($1 * 10^{-12}$ W)

The same noise source with the same sound power generates different sound pressure levels depending on the distance of the receiving point, on the position of the noise source (how close from a reflecting surface), on the environment conformation (more o less reverberating) and other factors. The sound power level (Lw), then, is an absolute quantity for each fan.
The sound pressure level (Lp), that is the quantity perceived by the user, depends on the installation, on the environment and on the position of the user against the fan.

ATTENTION: *The sound pressure level (Lp) indicated in every fan manufacturer catalogue is measured in particular testing conditions and may be different from what is obtained in a real installation.*

4.2 SOUND PRESSURE CALCULATION

Supposing a free field spherical propagation in an ideal elastic medium, the sound pressure level Lp, at a distance r from a noise source which sound power level is Lw, can be calculated with the following formula

$$Lp = Lw - 10 * \log 4 \pi {}^2 r + DI$$

or

$$Lp = Lw - 20 * \log r + DI - 11 \qquad [dB] \qquad (4.3)$$

where
*DI= 10 * log Q*

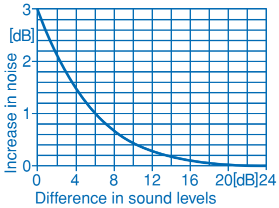
The term DI represents the effect of the source directionality, with Q defined as directionality factor.
The factor Q has different values, generally between 1 and 8. It is an empiric value, estimated according to experience. Some publications suggest an estimation of such value according to the position of the fan against reflecting walls, others according to the mutual position between the fan and the auditor in a certain environment.
Indeed it is often ignored, because the directionality effects of the sound source are covered by diffusion phenomena created by objects and surfaces present in the sound field.

Known the sound pressure level Lp1 in a certain point 1, located at distance r1 from noise source, the Lp2 level in point 2, at distance r2 in the same direction, can be calculated using the following formula

$$Lp_2 = Lp_1 + 20 * \log \frac{r_1}{r_2} [dB] \qquad (4.4)$$

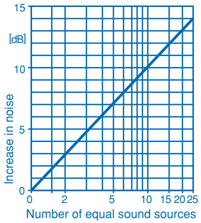
Example:
Lp1 = Sound pressure level at 3m = 60 dB
*Lp2 = Sound pressure level at 1m = 60 + 20*log3 = 69,5 dB*

In case of two sources with different noise levels, the combined value can be calculated using the following chart:



Example:
Two sources have sound values of 60 and 65 dB respectively.
The difference is 5 dB, that in the graph gives 1 as adding value.
The combined sound level is 65 + 1 = 66 dB

In case of various sources with the same sound level, the global value can be calculated using the following chart:



Example:
Five sources have a sound level of 60 dB
From the graph, the adding value is 7
The global sound level is 60 + 7 = 67 dB

4.3 SOUND SPECTRUM

Every sound or noise consists normally in a mix of sounds at different frequencies.
For better describing a sound, it is sometimes necessary to use sound octave bands (frequencies of 63, 125, 250, 500, 1000, 2000, 4000 and 8000 Hz are universally used), indicating for each frequency the level of sound power or pressure.

The human ear is more sensible to certain frequencies rather than others. For this reason the sound pressure level is often "weighted" (i.e. giving more weight to some frequencies rather than others), obtaining a total sound pressure value, starting from the values at each frequency, that better reflects the hearing sensation of the auditor. The "A" weighting scale, according to ISO 3744, is universally the one more often used and the calculated total value is normally indicated in dBA or dB(A).

4.4 FAN LAWS FOR NOISINESS

Also for fan noisiness there is a similitude law like the ones explained in Chapter 3. According to AMCA 300/67, there are the following relationships:

$$Lp_2 = Lp_1 + 50 * \log \frac{rpm_2}{rpm_1} \qquad (4.5)$$

$$Lp_2 = Lp_1 + 70 * \log \frac{D_2}{D_1} \qquad (4.5)$$

4.5 VIBRATIONS AND BALANCING

The vibrations generated by a fan are normally due to the residual unbalancing of the impellor and, for this reason, essentially of sinusoidal shape, with frequency equal to the rotational frequency of the impellor itself.
In time, it is possible that the impellor residual unbalancing can increase because of corrosion phenomena or, more commonly, because of dirt accumulation. Therefore it is essential, during periodical maintenance, to clean the impellor from such dirt deposits.

Other vibrations sources can be the presence of turbulences in the airflow (especially on fan inlet side), excessive pressure drop, natural frequency of the support structure too close to fan speed, problems on motor bearings…

Typical solution against vibrations transmission, from the fan to the support structure or to the duct system to which the fan is connected, is the use of anti-vibration supports (opportunely selected for the type of installation and for the weight of the fan) and of anti-vibration joints.

Vibrations are normally measured and described by the vibration speed V [mm/s]. In case of sinusoidal vibrations, the maximum speed of vibration is equal to the product $e\omega$ *where* e *is the residual eccentricity of the impellor baricenter and* ω *is the angular speed.*

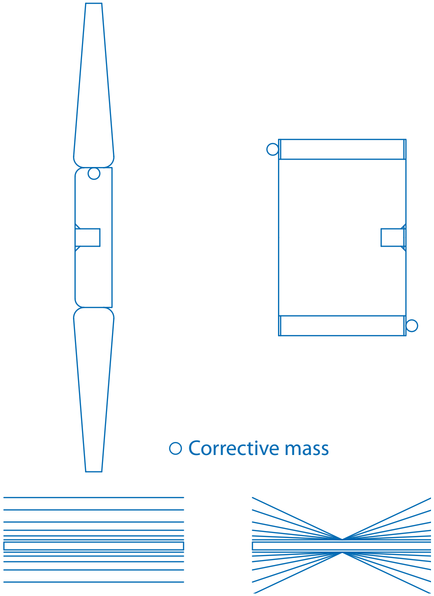
The norm ISO 1940 describes the acceptable vibrations level according to balancing degrees (G1 - G2,5 - G6,3 - etc.). The balancing degree normally approved for fans up to 15kW is G6,3. The degree of balancing points out the maximum speed of the vibrations.

For sinusoidal vibrations the effective value of the vibrations is equal to $1/\sqrt{2} * e\omega$*, or 0,71* $e\omega$*, therefore the effective value (rms) maximum acceptable is normally 4,5 mm/s.*

To minimize vibrations emission, impellors are subject to balancing, that can be either static or dynamic.
The static balancing process consists, in case an impellor of mass M has a heavier side (and so a residual eccentricity equal to e), in the addition of an additional weight m at a distance m diametrically opposite, on the same perpendicular plan to the impellor axis, such as

$$m * r = M * e$$

An impellor can be perfectly balanced from a static point of view, but still present a oscillating vibration in correspondence of its shaft.
This problem generally happens on impellors of a certain thickness (generally those of the centrifugal fans).
The dynamic balancing foresees the addition of additional masses on two different plans, both perpendicular to impellor axis, such to compensate the oscillating motion.



The balancing process can be made through rotating or non-rotating machines. The balancing process made through a rotating machine is frequently confused with the "dynamic" balancing, but the two things are completely different (being the real meaning of dynamic balancing explained above).

5 MOTORS AND DRIVES

Motors installed on Dynair fans are of different type, depending on fan typology, on application, on size… They can be shaded poles motors, external rotor motors, squirrel cage motors… In case of special applications, High Temperature or Explosion-proof motors are used. Please refer to specific catalogue section to know which motor type is installed on each fan range.

5.1 POWER SUPPLY

The rated power supply tension of the motors used by Dynair is generally 230V for single-phase motors and 230/400V or 400/690V for 3-phase motors. The frequency is 50Hz.
In case of different tension and/or frequency please refer to Dynair.

If the 3-phase power supply is 400V, 230/400V motors have to be connected in star and 400/690V motors have to be connected in delta.

5.2 STARTING

There are different methods for starting a motor.
The D.O.L. (Direct On Line) system, or direct starting, is the more commonly used. It has however the disadvantage to require an elevated starting current and this, especially in case of motors of big dimensions, can create problems to the power supply system and/or towards the energy supplying Company. For 400/690V 3-phase motors the most immediate solution is to use (if possible) a star/delta starting. This reduces both the starting current and torque down to approximately one third of the values that would occur in case of direct starting in delta. The connection in star must be done only at motor starting and for a brief period of time.

More sophisticated starting methods foresee the use of drives (inverter or soft-starter) that partialize supply frequency or tension at the start-up.

5.3 SPEED CONTROLLING

The need of controlling the performances of a ventilation system depends on various factors: comfort increase, noise emission reduction, adaptation to the environmental conditions, reduction of absorbed electric power...
The performances of a ventilation system can be easily controlled modifying the fan rotational speed.
The absorption of mechanical power to the fan shaft is reduced by the third power of the variation of rotational speed (if speed is halved, the power is reduced to an eighth of the absorption at full speed). By how much the absorbed electric power is reduced, however, it depends on the characteristics of motor and control drive.

There are different methods for controlling the speed of a fan. Below are listed the more common ones, that can be used with Dynair fans.

TENSION VARIATION

In small dimensions motors, speed can be regulated through the variation of the supply tension. This type of regulation is especially suitable for shaded pole and external rotor motors.
There are essentially two types of controllers: electronic and with autotransformer.
The ones with autotransformer are generally more expensive, but they increase the stability of the motor, being the tension supplied independent from the load. The tension regulation is not recommendable on 2-poles motors.

In the case of single-phase motor with capacitor, it is advisable (especially in case of use of an electronic controller) that the supply tension is varied only on the primary winding, leaving the secondary one (with capacitor) with full tension ("3-wires" connection, as described in the specific catalogue section).

FREQUENCY VARIATION

The speed of a squirrel cage motor can be effectively controlled varying the supply power frequency by mean of an inverter.
The way of speed controlling is ideal, although generally more expensive. So it is more frequently used on bigger size motors, in case of several motors controlled by the same inverter or in case it is required an extremely precise speed control.
Attention: not all motors can be controlled through inverter. It is recommended to ask advise to Dynair in advance.

As the motor cooling impeller is fitted, as standard, to the motor shaft, consequently also the cooling of the motor is reduced. On the other hand, also the power absorbed by the motor is reduced. In any case it is recommendable not to reduce the rotational speed below 50% of nominal speed (so not to reduce supply frequency below 25 Hz).
For motors of a certain size (and value…) it is appropriate to foresee for the motor a protection system against overheating (for example using PTCs).

If it is required to control a fan of small dimensions and mono-phase power supply is available, a valid solution (with good efficiency) is to use a small inverter with mono-phase inlet and three-phase outlet and a fan with three-phase motor.

DOUBLE POLARITY MOTORS

It is possible to install, on certain fan ranges, double polarity motors. Typically 2/4, 4/6, 4/8 and 6/8 poles motors are used, but not only. This solution allows to get different fan speed simply selecting the polarity of the motor itself.
4/6 e 6/8 poles motors have two separate windings.
2/4 e 4/8 poles motors can have a single winding (Dahlander, the most commonly used as it is cheaper) or two separate windings.

DELTA/STAR SWITCHING

Normally three-phase motors can be started in star connection and then quickly switched to delta connection. But they cannot run continuously in star connection.
Some specific Dynair fan ranges (indicated in our catalogue with "2V") are fitted with special increased slip motors and can be run continuously in star connection. The result is to have a double speed motor, similar to a 4/6 poles motor, but more economical.

VARIATION OF RAPPORTO DI TRASMISSIONE

It is possible only for belt driven fans and only for setting the system during installation and commissioning.
Changing ratio between pulleys diameters it is possible to have any required fan rotational speed and adapt very precisely fan performance to system requirements.

6 VENTILATION SYSTEM DESIGN

To design a ventilation system and to choose the appropriate fan and ducting (if necessary) type, it is necessary to take into account all the following elements:

- *type of application (industrial, commercial, domestic…)*
- *type of installation (air intake or supply free or ducted…)*
- *kind of carried fluid (clean air or with gas or dust, temperature, presence of explosive elements…)*
- *place of installation (wall, roof, false ceiling…) and eventual limitation of space and dimensions*
- *eventual noise level limits*
- *type of power supply (tension, frequency…)*
- *eventual required accessory (anti-vibration supports or joints, speed controller…)*

But, obviously, the most important parameters for the selection are the flow rate and, in case of ducted installation, pressure drop.

6.1 FLOW RATE

To design a ventilation system, the first step is to know the required air volume to be extracted or supplied from or to a certain environment in a certain period of time.

There several criteria to calculate such air volume. Frequently different criteria can be applied at the same time to a certain environment: in this case it is recommended to use the one indicating the higher air volume.
See below a short list of some of these criteria.

***N° OF AIR CHANGES PER HOUR RECOMMENDED FOR A CERTAIN ENVIRONMENT TYPOLOGY**

The airflow required (in m3/h) is given by environment volume (in m3) multiplied by recommended number of air changes per hour. See table below (note: values are indicative).

Environment	ric./ora
Chicken Farm	8÷15
Cattle Farm	15÷25
Hotel Halls	4
Garage (parking)	8
Garage (repairing)	10÷20
Lavatories - Showers	6
Galvanic Bath	25÷30
Banks	4
Cafes - Pubs	10
Carpentry	10÷12
Paper Hill	15÷20
Heating Plants	50÷60
Churches	10÷15
Cinemas / Theaters	10÷15
Dyers	15÷20
Tannery (drying)	35
Tannery (working)	18
Chromium Plating Plants	6÷10
Rubber Factories	10÷20
Bakeries	6÷10
Chemical Factories	15÷20
Environment	**ric./ora**
Factories (general)	6÷10
Woodworks	6÷15
Textile - Factoryes	5
Foundries	20÷30
Bread Oven	20÷30
Electric Ovens	30
Industrial Ovens	20
Furnace Rooms	20÷30
Mushroom Bed	10÷20

Halls	6÷20
Milk (working)	15
Cleaners - Dyers	20÷30
Boiler Houses (engine rooms)	20÷30
Warehouses for perishable goods	15
Warehouses for not perishable goods	5
Tobacco Processing	12
Canteens	4÷6
Motors (engine rooms)	5÷10
Mills	15÷30
Shops	5
Work Shops	6÷10
Hospitals	6
Gymnasium	10÷20
Swimming Pools	20÷30
Pump Rooms	6÷12
Restaurant (kitchens)	20÷40
Restaurant (rooms)	12
Environment	**ric./ora**
Waiting Rooms	10
Dancing Halls	8÷16
Casino	10÷20
Meeting Rooms	6÷8
Meeting Halls	10÷20
Schools	6
Dusty plants	10÷20
Plants metallurgik	5÷10
Supermarket	5÷10
Typography	15÷25
Toilet	30
Transformer Rooms	12÷30
Tecnical Rooms	15

***N° OF AIR CHANGES PER HOUR RECOMMENDED BY NUMBER OF PEOPLE IN THE ENVIRONMENT**

The airflow required (in m3/h) is given by the number of people normally present in a certain environment multiplied by the fresh air flow rate recommended by local norms (every Country has normally its own rules).
Generally the recommended fresh air flow rate is between 20 and 30 m3/h per person, increased by 10-20 m3/h in case smoke is allowed.

*** HEAT QUANTITY TO BE EXTRACTED FROM A CERTAIN ENVIRONMENT**

$$Q=\frac{P*3600}{\rho*cp*\Delta T} \qquad (6.1)$$

where:
Q = flow rate in m3/h
P = heat to be extracted in kW
ρ = air density in kg/m3
ΔT = temperature difference between air intake and outlet in °C
cp = air specific heat capacity (≈1)

6.2 PRESSURE DROP

In case of ducted ventilation, it is necessary to know the pressure drop given by the system.
A ventilation system is composed by various elements (ducting, bends, filters, grilles…) and has a static pressure drop given the sum of each element resistance.

The calculation of the system pressure drop is essential for a correct design and for a correct fan selection.
Pleas refer to specialized publications, dedicated software or specialist consultancy.

7 *CONVERSION FACTORS*

AIR FLOW								
Multiply	**for**	**to have**	**Multiply**	**for**	**to have**	**Multiply**	**for**	**to have**
	60	m^3/min		0,0003	m^3/s		0,000016	m^3/s
	3600	m^3/h		0,0167	m^3/min		0,001	m^3/min
m^3/s	1000	l/s	**m^3/h**	0,2778	l/s	**l/min**	0,06	m^3/h
	60000	l/min		16,667	l/min		0,0167	l/s
	2118,9	CFM		0,58858	CFM		0,03531	CFM
	0,0167	m^3/s		0,001	m^3/s		0,0004719	m^3/s
	60	m^3/h		0,06	m^3/min		0,02832	m^3/min
m^3/min	16,667	l/s	**l/s**	3,6	m^3/h	**CFM**	1,699	m^3/h
	1000	l/min		60	l/min		0,47195	l/s
	35,315	CFM		2,1189	CFM		28,317	l/min

PRESSURE								
Multiply	**for**	**to have**	**Multiply**	**for**	**to have**	**Multiply**	**for**	**to have**
	1	mmH_2O		0,10215	kgf/m^2		25,4	kgf/m^2
	0,07343	mmHg		0,10215	mmH_2O		25,4	mmH_2O
	9,7898	Pa		0,007501	mmHg		1,8651	mmHg
kg/m^2	0,0000966	Atm	**Pa**	0,0000099	Atm	**in-wg**	248,66	Pa
	0,00142	psi		0,000145	psi		0,002454	Atm
	0,03937	in-wg		0,004022	in-wg		0,03607	psi
	0,002891	in-Hg		0,0002953	in-Hg		0,07343	in-Hg
	1	kgf/m^2		10350	kgf/m^2		345,91	kgf/m^2
	0,07343	mmHg		10350	mmH_2O		345,91	mmH_2O
	9,7898	Pa		760	mmHg		25,4	mmHg
mmH_2O	0,0000966	Atm	**Atm**	101300	Pa	**in-Hg**	3386,4	Pa
	0,00142	psi		14,696	psi		0,03342	Atm
	0,03937	in-wg		407,48	in-wg		0,49115	psi
	0,002891	in-Hg		29,921	in-Hg		13,619	in-wg
	13,619	kgf/m^2		704,28	kgf/m^2			
	13,619	mmH_2O		704,28	mmH_2O			
	133,32	Pa		51,715	mmHg			
mmHg	0,001316	Atm	**psi**	6894,8	Pa			
	0,01934	psi		0,06805	Atm			
	0,53616	in-wg		27,728	in-wg			
	0,03937	in-Hg		2,036	in-Hg			

AIR FLOW								
Multiply	**for**	**to have**	**Multiply**	**for**	**to have**	**Multiply**	**for**	**to have**
m/s	60	m/min	**in/sec**	0,0254	m/s	**fpm**	0,00508	m/s
	39,37	in/sec		1,524	m/min		0,3048	m/mm
	3,2808	fps		0,0833	fps		0,2	in/sec
	196,85	fpm		5	fpm		0,0167	fps
m/min	0,0167	m/s	**fps**	0,3048	m/s			
	0,65617	in/sec		18,288	m/min			
	0,05468	fps		12	in/sec			
	3,2808	fpm		60	fpm			

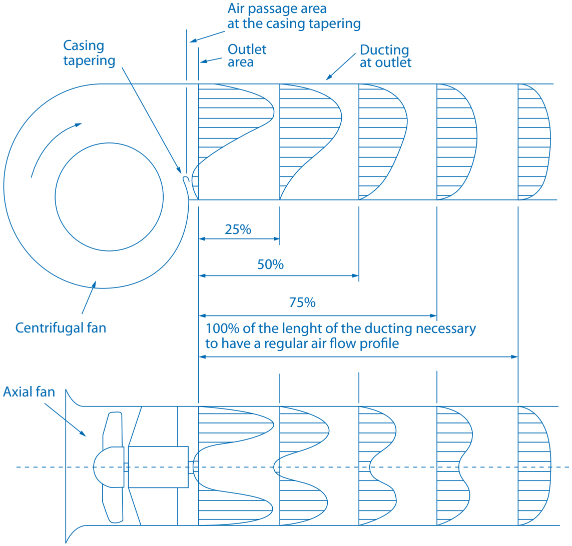
DENSITY								
Multiply	**for**	**to have**	**Multiply**	**for**	**to have**	**Multiply**	**for**	**to have**
kg/m^3	0,06243	lb/ft^3	**lb/ft^3**	16,02	kg/m^3			

8 INSTALLATION TIPS

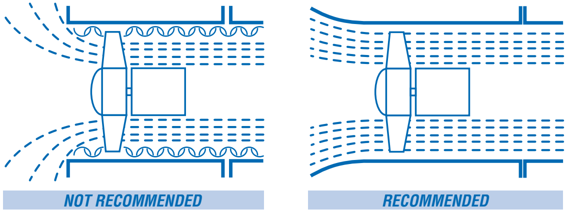
During design and installation of a ventilation system, there are some tricks that reduce the creation of turbulence. Turbulences unavoidably bring to a performance reduction, compared to what mentioned in the Catalogue (that is the result of laboratory test, made in ideal conditions according to precise reference norms) and to an increase of noise emission.

Find below a selection of these tips.

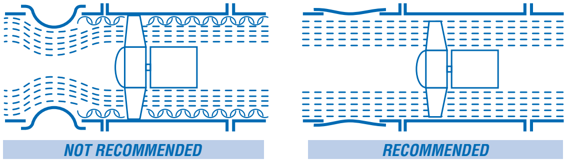
Before installing a “disturbing” element in the ducting (bend, split element, filter…) it is necessary to foresee, between the fan outlet and such element, a distance that allows the airflow to achieve a regular speed profile (see picture below). Such distance is generally equal to 2,5 times the duct diameter, for an average air speed lower than 12,5 m/s (in case of rectangular duct use equivalent diameter). Above such air speed value, it is necessary to add one diameter for each 5 m/s increase.

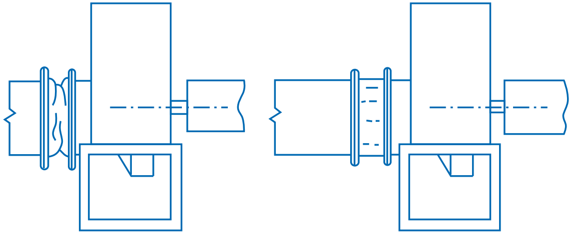


** In case of an axial fan installation with free inlet (not ducted) it is opportune to foresee an bell inlet cone.*



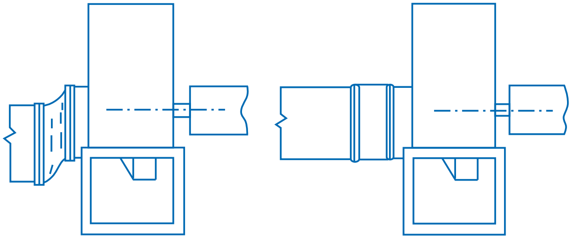
** In case of fan installation with anti-vibration joint, it is opportune that the same is installed reasonably tight.*





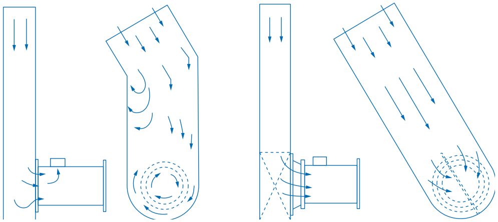
NOT RECOMMENDED | RECOMMENDED

* During installation it is opportune that ducting is well aligned with fan inlet (and outlet)



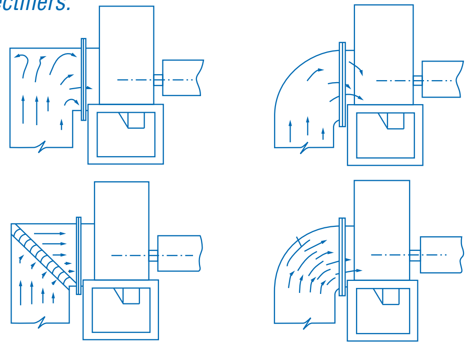
NOT RECOMMENDED | RECOMMENDED

* During installation it is opportune that ducting doesn't have a bend such to create air whirls at fan inlet.



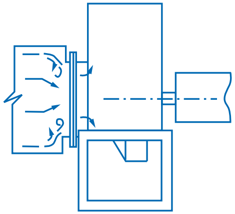
NOT RECOMMENDED | RECOMMENDED

* It is opportune that eventual bends close to fan inlet have air rectifiers.

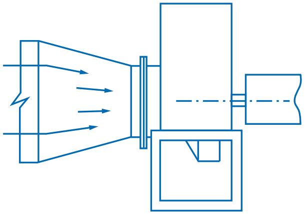


NOT RECOMMENDED | RECOMMENDED

** Eventual duct diameter changes, especially if close to fan inlet, must be smooth, possibly with small angle.*

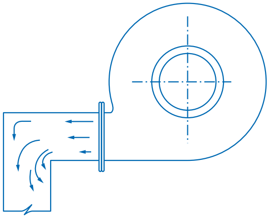


NOT RECOMMENDED

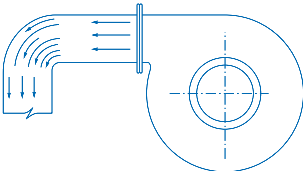


RECOMMENDED

** If it necessary to install a bend close to fan outlet, it is necessary for the bend to be smooth and with air rectifiers.*

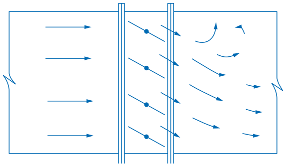


NOT RECOMMENDED

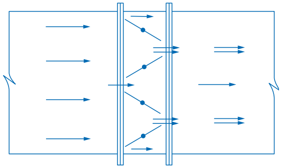


RECOMMENDED

** If it is necessary to install setting shutters along the duct, it is preferable to use shutters with opposite blades, to reduce creation of whirls and turbulence.*



NOT RECOMMENDED



RECOMMENDED

9 CONCLUSIONS

This section of the present catalogue has a divulgative purpose . Dynair cannot be held responsible in case of possible inaccuracies, errors and/or omissions.

According with our policy of continuous development, all data included in Dynair catalogue can be modified without prior or successive notice

Dynair remains at your disposal for consultancies, fan selections and offers.