

# > Principal technical particulars <

## 2.1 FLOW RATE DEFINITION

The flow rate in volume is the fluid volume passing through the fan in the time unit.

In less frequent use is the flow rate in weight, defined as the fluid weight passing through the fan in the time unit.

Knowing the flow rate of a fan, linked to a duct, one can calculate the speed of the fluid in the duct section with the formula:

$$v = \frac{Qv}{3.600 \cdot A} \quad (2.1)$$

where:

$v$  = Average fluid speed in m/s

$Qv$  = Flow rate in m<sup>3</sup>/h

$A$  = Area of the duct section in m<sup>2</sup>

Vice versa, knowing the speed  $v$  of the fluid and the section area  $A$  of the duct, the flow rate can be calculated:

$$Qv = v \cdot A \cdot 3600 \quad (2.2)$$

where:

$Qv$  = Flow rate in m<sup>3</sup>/h

$v$  = Average speed of the fluid in m/s

$A$  = Area of the duct section in m<sup>2</sup>

## 2.2 TOTAL, STATIC AND VELOCITY PRESSURE DEFINITION

When moving fluid three types of pressure can be distinguished.

### 1 - Static pressure (ps)

This is defined as the pressure of the fluid exercised on the walls of the duct or the recipient where it is contained.

This acts in all directions and is independent of fluid speed. Taking ambient pressure as reference, the static pressure is positive when it is greater than ambient pressure, and negative when it is lower.

### 2 - Velocity pressure (pd)

This is defined as the pressure corresponding to the energy part contained in the fluid mass unit because of its speed (kinetic energy).

This acts in the same direction as the fluid moves and is always considered positive.

The velocity pressure is function of the fluid density and speed and is expressed by the following formula:

$$pd = \frac{1}{2} \cdot \rho \cdot v^2 \quad (2.3)$$

where:

$Pd$  = Velocity pressure in Pa (Pascal)

$\rho$  = Fluid density in Kg/m<sup>3</sup>

$v$  = Fluid velocity in m/s

The velocity pressure, expressed in mm H<sub>2</sub>O, can be calculated with reasonable approximation, for air in normal technical conditions, by the following practical formula:

$$pd = \frac{v^2}{16} \quad (2.4)$$

where:

$Pd$  = Velocity pressure in mm H<sub>2</sub>O (da Pa)

$v$  = Fluid velocity in m/s

### 3 - Total pressure (pt)

This is defined as the algebraic sum of the static (ps) and velocity (pd) pressure:

$$pt = ps + pd \quad (2.5)$$

Special working conditions of the fan are:

- a) working with closed inlet and / or outlet (shut-off delivery)
- b) working with inlet and out let not attached to ducts (free delivery)

Working in shut-off delivery, the flow rate is zero. Therefore the fluid speed and the velocity pressure are zero.  
In the case:

$$pt = ps$$

This is working condition corresponds to the starting point of the fan performance curve. The fan is freeflow when neither the inlet or outlet are linked to ducts. In this case the static pressure will be zero so:

$$pt = pd$$

The pressure generated by the fan is all kinetic and, this working condition corresponds to the final point of the fan performance curve.

### 2.3 PRESSURE MEASUREMENTS

The static pressure is measured by connecting a manometer (for example a water manometer) between the wall of the duct and the external ambience (see fig. 1 and 2).

The velocity pressure on the contrary, is measured by connecting a manometer between the duct wall and an internal point by means of a tube with the terminal opening exactly opposite to the flow direction (see fig. 1 and 2).

The total pressure, finally is measured by linking a manometer between the external environment and an internal point by means of a tube with the terminal opening exactly opposite to the flow direction (see fig. 1 and 2).

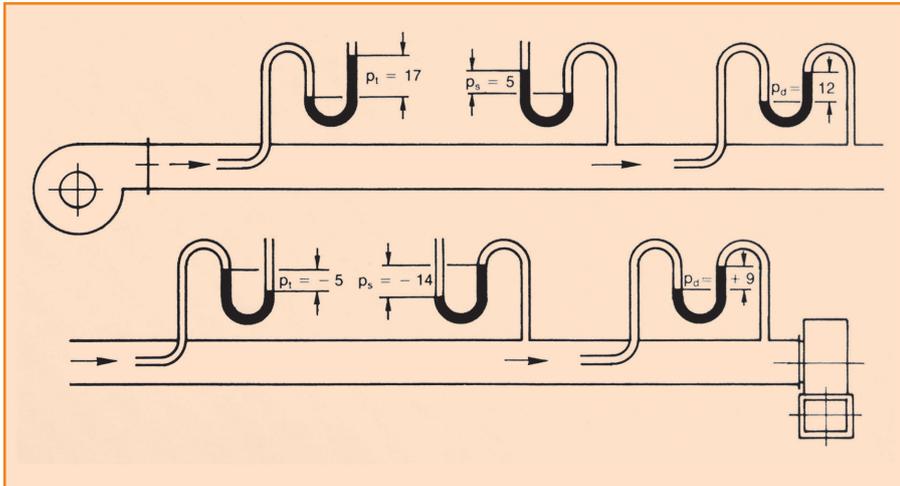


Fig. 1 - Fan with outlet connected to the duct. The pressure into the duct is bigger than ambient pressure.

Total pressure - Static pressure = Velocity pressure

$$pt (+ 17) - ps (+5) = pd (+ 12)$$

Fig. 2 - Fan with inlet connected to the duct. The pressure into the duct is lower than ambient pressure.

Total pressure - Static pressure = Velocity pressure

$$pt (- 5) - ps (- 14) = pd (+ 9)$$

### 2.4 SHAFT POWER DEFINITION

In order to supply a flow rate, with a fixed total pressure, a fan requires some mechanical power supplied by an electric motor. This power, depending also on the fan efficiency, is given by the following formula:

$$Pv = \frac{Qv \cdot pt}{3,671 \cdot \eta} \quad (2.6)$$

where :

$Pv$  = Shaft power in W

$Qv$  = Flow rate in  $m^3/h$

$Pt$  = Total pressure in mm  $H_2O$

$\eta$  = efficiency in %

## 2.5 FAN CHARACTERISTIC CURVE

The energy a fan receives from the electric motor is transferred to the fluid, passing through it in the form of total pressure (pt).

The total pressure a fan can supply is not constant, but varies in relation to the flow rate.

Also the power requirement varies in relation to the flow rate.

For the practical use of the fan it is necessary to know the total pressure available and mechanical power requirement for each flow rate value.

This information can be given, for each fan and for a fixed number of revolutions in the form of a graph, in Cartesian coordinates where in abscissas the flow rate values and in ordinates the pressures and powers are shown (or in the form of tables),

In the catalogue the characteristic curves are published (for the belt drive fans) that show the total pressure (Pt), dynamic pressure (db) shaft power (Pv) in relation of flow rate (Qv) and the revolution number (r.p.m.); the efficiency values ( $\eta$ ) and noise values (Lp, Lw) are shown.

Concerning the direct drive arrangements, some tables are also published and they show the total pressure (pt), the flow rate (Qv), installed and shaft powers (Pv), the revolution number (r.p.m.) and the noise (Lp).

## 2.6 CHARACTERISTIC CURVE OF THE SYSTEM

If you want some quantity of air to circulate in a duct system it is necessary to supply some energy to the fluid, in form of pressure, to be able to overcome the resistance in movement.

The pressure to be supplied varies with the flow rate and the relationship existing between these two quantities is given by the following expression:

$$p = K \cdot Qv^2 \quad (2.7)$$

where:

p = Pressure

K = coefficient depending on the characteristic of the system (fluid movement resistance)

Qv = Flow rate

If the characteristic of the system do not vary, the coefficient K remains constant for not very wide variations in the flow rate and can be calculated from the formula:

$$K = \frac{p}{Qv^2} \quad (2.8)$$

Calculating the K, for one of the working conditions, it is possible to trace a curve of the p in relation to the Qv, that is the "system characteristic curve".

In Cartesian coordinated and with linear scale, this curve is a parabola passing from the origin of the axis (see fig. 3).

## 2.7 WORKING POINT

A fan, installed in a system, will supply a flow rate corresponding to the value of the static pressure necessary to overcome the resistance to the fluid movement in the circuit.

Laying out both the fan static pressure curve and the system fluid movement resistance curve in the same diagram, the crossing point of the two curves will be the "working point" (point A of fig. 3).



**Better use zone**

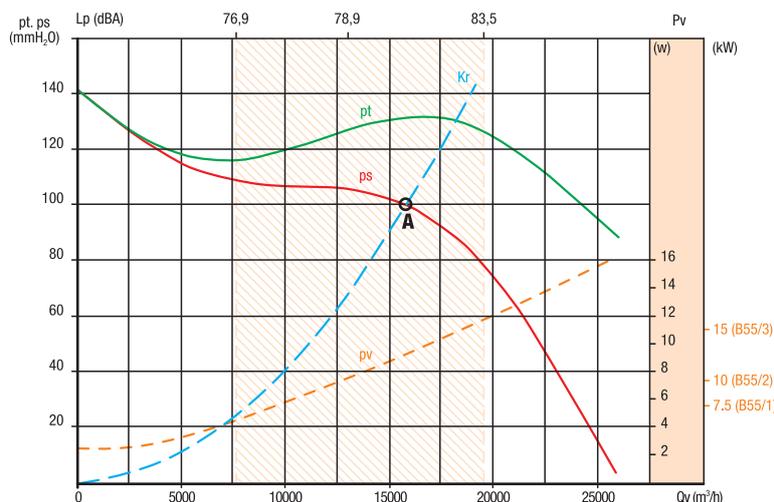


fig. 3

The working point is a point of equilibrium, as, in these conditions, the static pressure supplied by the fan equals the one required by the system to overcome the fluid movement resistance.

If the resistance of the circuit varies, for example with the partial closing or opening of a shutter, or because of the insertion of further elbows curves, a new characteristic system curve can be traced and a new equilibrium point be found with the fan static pressure curve.

Then the resulting variation of the flow rate can be valued.

It is advisable that the working point of the fan be always within the better use zone (central area of the characteristic curves) because the efficiency of the fan is higher in this area. This allows a saving of the absorbed power and lower noise level under the same performances.

It is advisable not to operate the fan in a point of the curve too much on the right (the area near to free delivery), as they would have too high velocity pressures, low efficiency and high noise level.

*In the case it is more suitable to choose a bigger model in the same series.*

It is also advisable not to choose a point of the curve too much on the left (area near the shut off delivery) because the efficiency would be too low and the working unstable.

*In this case it is more suitable to choose a smaller fan of the same series.*

## 2.8 MEASUREMENT AND CONVERSION UNITS

For the benefit of users, the definitions of the principal measurement units used in the ventilation technique and conversion ratios among the equivalent units are shown here.

For the flow rate in volume the international System (SI) adopts the m<sup>3</sup>/s. However, as already said, in practice, the m<sup>3</sup>/h is used more frequently and other different unit of measure. In English countries the cubic foot per minute (cfm).

The ratio existing between these two units is:

measurement unit	m <sup>3</sup> /s	m <sup>3</sup> /h	m <sup>3</sup> /1'	l/s	l/1'	cfm
1 m <sup>3</sup> /s	1	3600	60	1000	60000	2119
1 m <sup>3</sup> /h	0,0002777	1	0,016666	0,2777	16,666	5,8857
1 m <sup>3</sup> /1'	0,016666	60	1	16,666	1000	35,314
1 l/s	0,001	3,6	0,06	1	60	2,12
1 l/1'	1,666x10 <sup>-5</sup>	0,06	0,001	0,016666	1	0,03531
1 cfm	0,000472	1,7	0,02831	0,472	28,3168	1

The pressure for SI is the Pascal (Pa), corresponding to the force of 1 newton on the surface of 1m<sup>2</sup>.

The units frequently used on the contrary are: the mm H<sub>2</sub>O, corresponding to the force of 1kilogram on the surface of 1m<sup>2</sup> (1 mm H<sub>2</sub>O = 1 kgf/m<sup>2</sup>), the torr (mm Kg), the bar and millibar (mbar).

The English pressure unit is the inch water gage (inwg).

measurement unit	Pa	mm H <sub>2</sub> O (kgf/m <sup>2</sup> )	torr (mm Hg)	bar	mbar	inwg
1 Pa (N/m <sup>2</sup> )	1	0,102	0,0075	0,00001	0,01	0,004
1 mm H <sub>2</sub> O (kgf/m <sup>2</sup> )	9,806	1	0,0735	9,806x10 <sup>-5</sup>	0,098	0,0393
torr (mm Hg)	133,32	13,6	1	0,0013	1,3332	0,5352
bar	100000	10197	750,06	1	1000	401,46
mbar	100	10,2	0,75	0,001	1	0,4014
inwg	249	25,4	1,8683	0,0024	2,49	1

For the velocity, the unit adopted by SI is the m/s and is adopted in the ventilation technique too.

The English velocity unit is the foot per minute. The conversion formula is:

$$1 \text{ fpm (ft/m)} = 0,00508 \text{ m/s}$$

For the temperature, SI adopts two measurements units: degree Celsius (°C) and degree Kelvin (°K).

The ratio between the two units is:

$$°K = 273,15 + °C$$

The English temperature unit is the degree Fahrenheit (°F) and the conversion formula is:

$$°F = \frac{9}{5} \cdot °C + 32$$

$$°C = \frac{5}{9} \cdot (°F - 32)$$

The power unit in the SI is the watt (W) or kilowatt (kW)

measurement unit	W	kW	HP	CV	kcal/h
1 W	1	0,001	0,00134	0,00136	0,86
1 kW	1000	1	1,341	1,36	860
1 HP	745,7	0,7457	1	1,014	641,18
1 CV	735,5	0,7355	0,986	1	632,35
kcal/h	1,162	0,00116	0,00156	0,001582	1

The ratio existing among other English units in more frequent use and the SI corresponding ones are:

measurement unit	mm (millimetro)	m (metro)	ft (piede)	in (pollice)
1 mm (millimetro)	1	0,001	0,0033	0,0396
1 m (metro)	1000	1	3,28	39,37
1 ft (piede)	304,8	0,3048	1	12
1 in (pollice)	25,4	0,0254	0,0833	1

measurement unit	l (litro)	m <sup>3</sup> (metro cubico)	cuft (piede cubico)	cuin (pollice cubico)
1 l (litro)	1	0,001	0,0353	61
1 m <sup>3</sup> (metro cubo)	1000	1	35,314	61023,74
1 cuft (piede cubico)	28,32	0,02832	1	1728
1 cuin (pollice cubico)	0,0163	1,63x10 <sup>-5</sup>	0,00057	1

1 lb (libbra) = 0,4536 kg

1 kg = 2,2045 lb

## 2.9 FAN CLASSIFICATION AND TERMINOLOGY

The classification of the various types of fans and the relative terminology were standardized by UNI 7972 standards.

Above all, in these standards the term “fan” is defined. This term indicates the general working machine, without any additional element either at the inlet or outlet.

The term “extractor” refers to particular machine, that is a machine prepared for connection to piping only from the inlet section (e.g. roof extractor).

With regards to machine performances the following classification is given:

**a) Low pressure fans:** fans for pressure under 720 pascal (< 73 mm H<sub>2</sub>O).

**b) Medium pressure fans:** fans for pressure from 720 to 3600 pascal (73 ÷ 367 mm H<sub>2</sub>O).

**c) High-pressure fans:** fans for pressure above 3600 pascal (> 367 mm H<sub>2</sub>O).

With regards to working conditions, the UNI standards show the following:

### a) Normal use fans:

Suitable fan for non-toxic, non corrosive, unflammable air, without liquid or solid particles, with a maximum temperature of 80°C, or 40°C in case the motor and the fan supports are crossed by the conveyed air.

### b) Special use fans:

- **Hot gas fan:** suitable for treating hot gases, whose temperature must be within specified values. This must be manufactured with high temperature resistant material and can be supplied with a support cooler.

- **Biphase fluid (gas/liquid) fan:** suitable for conveying air containing liquid particles.

This fan can be equipped with a device for the liquid drainage and protections or materials suitable against corrosion and erosion.

- **Gas tight fan:** suitable to reduce the leakage of the conveyed gas and/or the entry of external air.

The degree of tightness depends on the type and pressure of the conveyed gas.

- **Dust transport fan:** suitable for extracting air containing dust and designed for the special type of the conveyed dust.

- **Crossed fan for solid material transport:** suitable for extracting air containing solid materials (e.g. wood chips, textile fibers, powdery material) and it is designed for special type of the material conveyed.

- **Antideposit fan:** designed to reduce to a minimum the depositing of the transported material and it is equipped with device for its periodic cleaning.

- **Abrasion proof fan:** designed to reduce abrasion to a minimum.

The parts subject to wear out are manufactured in material suitable for abrasion and /or can be easily replaced.

- **Corrosion proof fan:** manufactured or coated with suitable material to resist corrosion from the conveyed fluid.

- **Antispark fan:** designed to reduce the risk of sparks resulting from the rubbing of its component parts or with external material. Different types of construction are available, depending on the safety precautions considered necessary and defined by new ATEX standards.

- **Roof fan (roof-extractor):** designed to be installed on the roof and protected against atmospheric agents.

The above listed categories of special applications, are the most typical and frequent cases in the ventilation field.

There are, besides these, other types of fans with special characteristics.

The fans can obviously gather more than one of the described characteristics.

## 2.10 BELT DRIVE FANS

Sometimes in industrial installations or in ventilation plants some pressure and flow rate performances are required, with a very low level of tolerance. In such cases it is difficult to satisfy the request using a direct drive fans.

One model might offer insufficient performances while the next in the same series might have too high performances and be not acceptable for this reason.

On the other hand the manufacturing of a special model with a suitable impeller diameter, between two consecutive impeller diameters available in the series is a solution not economic. In this case we prefer to adjust the performances of the standard fan by varying the number of revolutions.

As it is stated in paragraph [4.3], the asynchronous motor rotation speed cannot be easily and conveniently varied. Thus, to vary the speed of the impeller, it is necessary to leave the direct drive type and to apply to the belt drive model.

With belt driving it is possible to obtain the required impeller speed, once the speed of the motor has been chosen; varying the ratio between the diameters of the pulleys according to the well known formula:

$$\frac{n_g}{n_m} = \frac{D_m}{D_g} \quad (2.9)$$

where:

$n_g$  = number of revolutions of the impeller

$n_m$  = number of revolutions of the motore

$D_m$  = diameter of the pulley fitted to the motor shaft

$D_g$  = diameter of the pulley fitted to the impeller shaft

Example (see page 41):

$Q_v = 42.000 \text{ m}^3/\text{h}$

$P_s = 1.600 \text{ Pa}$  (163 mm  $\text{H}_2\text{O}$ )

$t = +20^\circ\text{C}$  (working)

a) Read in abscissa, in correspondence of requested flow rate, the corresponding dynamic pressure (pd); in this case:

$p_d = 400 \text{ Pa}$

Adding up pd to requested ps to obtain pt.

$p_t = p_s + p_d = 1.600 + 400 = 2.000 \text{ Pa}$

b) Select in graphic the working point Flow rate/Pressure corresponding to the obtained values.

Now we can have the following data:

c) Number of revolution = 1.650 r.p.m. (revolutions of impeller)

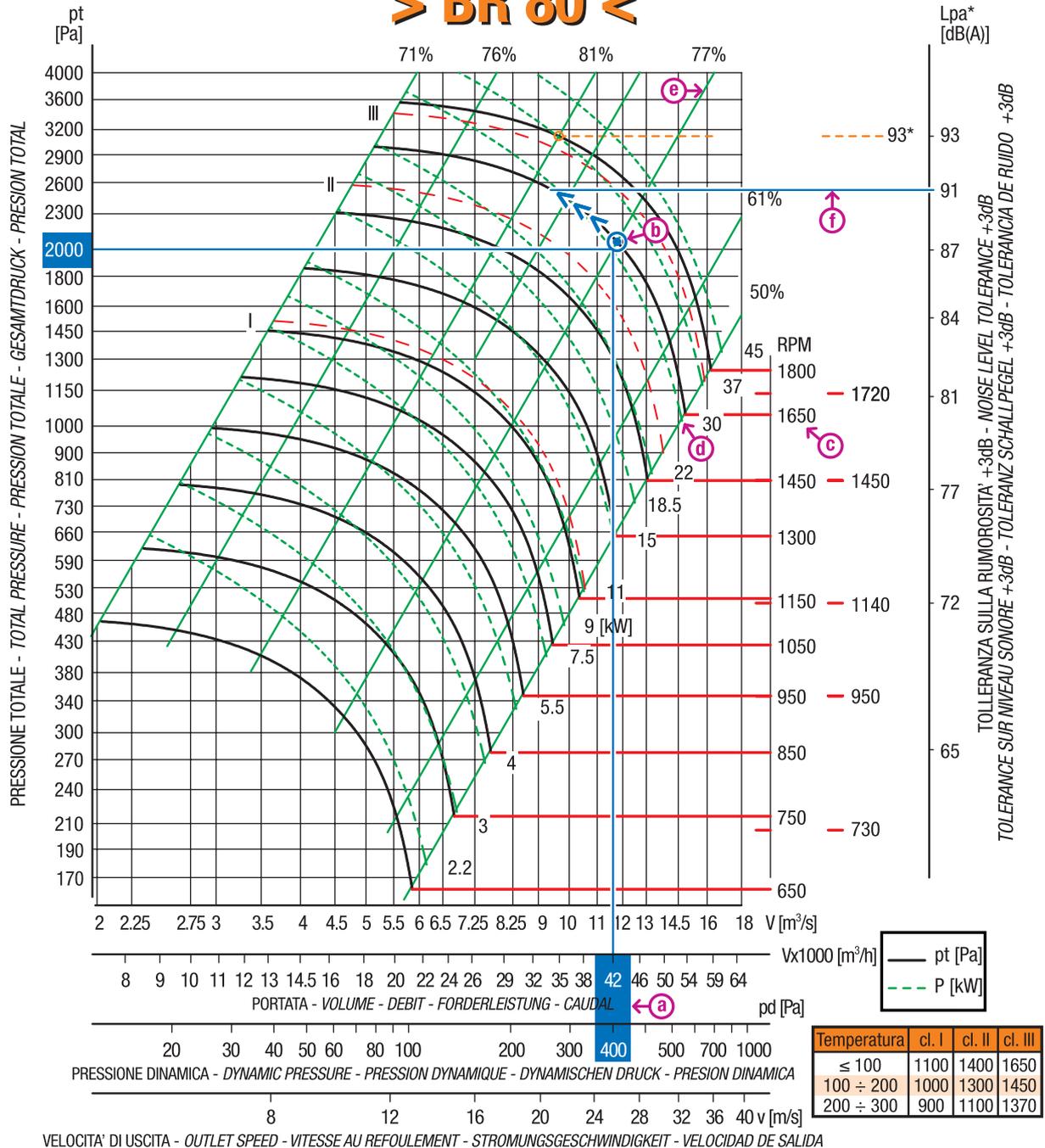
d) Shaft power = 30 kW (green dashed curve)

e)  $\eta = 77\%$  (green diagonal line)

f)  $L_p = 91 \text{ dB(A)}$ .

Running through the flow rate/Pressure curve, from the working point, to cross the diagonal of maximum efficiency ( $\eta = 81\%$ ) and so to move horizontally towards right until you cross the sound pressure scale LP.

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VELOCITA' DI USCITA - OUTLET SPEED - VITESSE AU REFOULEMENT - STROMUNGSGESCHWINDIGKEIT - VELOCIDAD DE SALIDA

Livello potenza sonora - Sound power level - Niveau de puissance sonore - Schalleistungspegel - Nivel potencia Sonora [dB] *									
RPM	Banda d'ottava - Octave band - Bande d'octave - Oktavband - Banda de octava [Hz]								Lpa* dB(A)
	63	125	250	500	1000	2000	4000	8000	
1080	94.9	94.9	96.9	91.9	90.9	85.9	77.9	77.9	81
1260	98.3	98.3	100.3	95.3	94.3	89.3	81.3	81.3	84.4
1440	101.3	101.3	103.3	98.3	97.3	92.3	84.3	84.3	87.3
1620	103.9	103.9	102.9	103.9	99.9	94.9	86.9	86.9	90.7
1800	106.2	106.2	105.2	106.2	102.2	97.2	89.2	89.2	93

[\*] Al massimo rendimento - At max. efficiency - A la maxime efficacité - zur höchsten Leistung - Al Máximo rendimiento - Distanza - Distance - Distance - Abstand - Distancia : 1,5 m - Temperatura - Temperature - Température - Temperatur - Temperatura : 15 [°] - Densità - Density - Densité - Bezugsdichte - Densidad : 1,225 [kg/m³] - Tolleranza sulla portata - Capacity tolerance - Tolérance sur le débit - Fordertoleranz - Tolerancia caudal : ±5%  
 PD² - WD² - GD² - PD² : 26 kgm²

### Summing up the data:

$Q_v = 42.000 \text{ m}^3/\text{h}$

$P_t = 2000 \text{ Pa}$  ( $p_s = 1600 \text{ Pa}$ ;  $p_d = 400 \text{ Pa}$ )

Shaft power = 30 kW (install 37 kW motor)

$\eta = 77\%$

$L_p = 91 \text{ dB(A)}$

Number of revolution = 1.650 r.p.m. (revolutions of impeller)

Wanting to install a 37 kW motor 4 poles 50 Hz ( $\approx 1.475 \text{ r.p.m.}$ ) the belt ratio to give to the diameters of pulley will be:

$R_t = \text{r.p.m. motor} / \text{r.p.m. impeller} = 1.475 / 1.650 = 0,89$