Product description
Design conditions

General information
All fans listed herein are single-stage fans and manufactured in welded industrial design. Depending on the fan design, there is a common base frame made of steel to accommodate the housing, bearings and drive motor. The impellers are designed in accordance with state-of-the-art flow technology. The performance data is based on an inlet temperature of 20° C at an air pressure of 101,325 Pa relative to a density of 1.205 kg/m³. The presently effective EU Directives are the basis for fan and equipment design. The technical design complies with DIN 24166, „Technical Delivery Conditions for Fans“. We are certified according to DIN EN ISO 9001.

Ambient temperature
The mechanical drive elements are designed for steady ambient temperatures of -10°C up to +40°C. Please consult us for other temperatures.

Operating temperature
All fan components in contact with the medium are designed to handle temperature ranges from -10°C to +80°C. The fan types MX, KX and RG are suitable for temperatures of up to +180°C provided a heat flinger is installed. Please consult us for other temperatures.

Balancing technique
All fan impellers and other rotating parts are dynamically balanced in two planes. The permissible residual unbalance has been determined in compliance with the Standard DIN ISO 1940 Part I „Balance quality requirements of rigid rotors“. The balance quality grade of the entire fan unit is G 6.3. Other quality grades can be supplied for special requirements. Please enquire about the additional costs for other quality grades. All connections with key and slot are balanced by half keys according to DIN EN 60034-14.

Vibration technique
All fans meet the technical vibration requirements with respect to dynamic rigidity, quiet running, noise radiation and similar aspects. The evaluation of the vibration complies with DIN ISO 10816-3 „Evaluation of machine vibration by measurements on non-rotating parts – part 3: Industrial machines with nominal power above 15 kW and nominal speeds between 120 r/min and 15,000 r/min when measured in situ (ISO 10916-3:2009)“. Other applicable regulations: DIN ISO 10816 „Evaluation of machine vibration by measurements on non-rotating parts“ and DIN ISO 13373-1 “Condition monitoring and diagnostics of machines“ Part 1: General procedures“.
Design conditions

Mechanical load
The fan design must ensure that all fan components can handle all possible load situations which may occur. Impellers in particular are key components heavily loaded. Apart from the static stress to which they are subjected, impellers also have to cope with supplementary dynamic loads. Frequent load changes caused by permanent control intervals and the aerodynamic influence of other plant components may substantially shorten the service life of the impeller and other components subjected to mechanical loads. Therefore it must be ensured that load changes are reduced. Load changes can have different causes:

a) Speed-dependent excitation
   e.g. frequent start from standstill, control with a frequency converter, pole-changing motor etc.

b) Aerodynamic excitation
   provided that the entire system resonates and is subjected to an unsteady volume flow or to deviation in pressure caused by unstable operating points (e.g. distinctive pumping of the system, constant change of operating points etc.)

c) Vibration-dependent excitation  e.g. caused by vibrations of the complete shafting system (motor, coupling, fan, shaft, impeller etc.); see standard VDI 3840 „Vibration of shafting systems“

In order to keep impact loads and alternating loads acting on mechanical components as small as possible, frequent switching must be avoided. High switching frequencies always occur whenever there are erratic changes of speed (e.g. starting from shutdown, speed changes of pole-changing motors and permanent control intervals by frequency converters, etc.). It must therefore be ensured that switching frequencies of 6 to 8 times per day are not exceeded. Please consult us for higher switching frequencies.

Remarks on transport
Fans should always be transported using means of transport that are suitable for the location where the fan is to be installed. Only use the lifting lugs and a fork-lift truck to lift and transport the fan. Do not attach lifting tackles to inlet or discharge or to the fan motor. Blank off all openings (nozzles, flanges, etc.) firmly to keep out foreign substances, moisture, dust, etc.

Guarantee claims
We reserve the right to change all technical data shown in this list. Guarantee claims made as a result of such technical modifications are excluded. Prior to mounting and commissioning on-site, the corresponding safety instructions and operating manuals have to be read. Adequate preservative measures are required for long-term storage on-site. Storage instructions are available on request.
Design conditions

Information required from the customer

Where the customer’s order does not refer to catalogue details, the following information is required to select the correct type of fan and the required equipment.

1. Ambient conditions (e.g. ambient pressure or altitude, ambient temperature and relative humidity)
2. Inlet temperature and inlet density
3. Total pressure increase
4. Volume flow (based on the inlet condition at temperature \( t_1 \) and static pressure \( p_{st1} \))
5. Handled gas or type of gas and its composition (gas constant); details as to whether the gas is e.g. explosive, aggressive, corrosive, dust or moisture containing, toxic or radioactive; type, composition and grain size distribution of the dust contained in the medium if the dust content of the gas handled is considerably higher than that of the external air in industrial areas (> 5 mg/m³) (e.g. abrasive, adhesive, sticking and hygroscopic)
6. Information on the type of the system, machine or unit and its intended use; mounting and installation conditions, installation dimensions to be kept
7. Type of installation and connection:
   - A, B, C or D, see page PB 16.
8. Operating conditions such as permanent operation, interrupted operation, long standstill periods, start-up frequency and variable speed control by frequency control.
9. Type of drive, method of switching-on of the motor
   - see pages PB 13 to PB 15.
10. Voltage, frequency, special conditions of the supply system
    - Please pay particular attention to the speed changes and the resulting changes in output for a 60 Hz mains.
    - Please request further information from us.
11. Design features such as e.g. radial fan, position of housing, type of drive (e.g. belt or coupling) and arrangement or control units, if necessary.
    Please note:
    The fan sense of rotation is viewed from the fan driven end in the direction of the motor fan impeller (generally, this is the non-driven end motor bearing). The motor sense of rotation, however, is viewed from the free motor shaft end (usually, this is the driven end motor bearing), that is to say, the view direction is from the opposite side of the motor fan impeller. If, for example, the fan sense of rotation is clockwise, the motor sense of rotation is opposed to that, namely counter-clockwise.
12. Whether sealing is required for the housing and the shaft passage and whether gas has to be prevented from entering or escaping.
13. Other general information (e.g. protection against corrosion, information on material, life time of bearings, ducting forces, earthquake and vibration load, pressure-resistant and shockproof or gastight). Equipment e.g. guards, flexible connections, control devices and dampers, suction boxes.

Special note on guarantee

In our capacity as the fan supplier, we are not generally familiar with the individual system design and local conditions. Therefore, the system designer or project manager should prepare the ordering data in such a way that it already refers to the peculiarities and local conditions that are specific to the system. Our guarantee details refer to individual values and test conditions in accordance with the presently effective DIN Guidelines or Standards and Regulations. It is therefore essential to adapt these to the operating state under given local conditions. For built-in parts belonging to our scope of delivery such as transition pieces, dampers, suction boxes, filters, guards, silencers etc the system designer is required to determine the corresponding installation resistances and include them in the calculations which are required to determine the fan type.
Structural designs

Type designation and article number
KX E 040 - 0500 15 - 00

internal code
impeller speed \( n_L \cdot 100 \text{ [min}^{-1}] \) e.g.: 1,500 min\(^{-1}\)
volume flow \( V_L \text{ [m}^3\text{/min]} \) e.g.: 500 m³/min
total pressure increase \( \Delta p_L \cdot 10 \text{ [daPa]} \) e.g.: 400 daPa

\( E = \) single-stage radial fan

structural design: MX motor on pedestal; MA flange-mounted motor; KX with coupling; RG with belt drive

Important data needed to handle your order

1. Structural designs
   - MXE
   - MAE
   - KXE
   - RGE
   description see page PB 6

2. Sense of rotation
   - counterclockwise rotation
   - clockwise rotation
   (viewed from the driven end) see page PB 7

3. Position of discharge

4. Arrangement of the motor
   - right side
   - left side
   (for structural design RGE only)
   see page PB 8

5. Motor size (if provided by customer)
   \( P_M = \) _______ kW
   \( n_M = \) _______ min\(^{-1}\)
   motor size = _______ make = _______

6. Placing of terminal box
   - left side
   - on top
   - right side
   viewed towards the motor shaft end
Structural designs

Structural design MXE
Directly driven by the motor shaft on which the impeller is mounted. The motor of foot mounting type (IMB3) is placed on the pedestal.

Structural design MAE
Directly driven by the motor shaft on which the impeller is mounted, suitable for surface mounting on a separator, filter etc. The motor of flange design (IMB5, IMV1) is directly flanged to the fan housing.

Structural design KXE
Power transmission from motor shaft to fan shaft by a flexible coupling. The fan shaft runs in two antifriction bearings.

Structural design RGE
Power transmission from motor shaft to fan shaft by V-belts. The fan shaft runs in two antifriction bearings. The motor is laterally arranged on a base frame made of channel.
Design options

Sense of rotation
Single-stage radial fans are available in two directions of rotation. As viewed from the driven end it is:

- **GR** = clockwise rotation [RD]*
- **GL** = counterclockwise rotation [LG]*

* Identification in [......] as per EUROVENT

Positions of discharge
The position of the housing or the direction of the discharge is indicated in position degrees. The sense of rotation or sense of impeller rotation is always indicated as viewed from the driven end VDMA 24 165.

![Diagram of sense of rotation and positions of discharge](image-url)
Motor arrangement

For structural design "RG" the motor can be arranged at the right or at the left side.

**Design r**
Arrangement of the motor on the right side of the base frame, viewed from the driven end.

**Design l**
Arrangement of the motor on the left side of the base frame, viewed from the driven end.

**Design variants**
Arrangement of the motor and sense of rotation for structural design RG
Position of the inspection openings (IO)

Example 1: Discharge position GR [RD] 360°
IO position at 45°

Example 2: Discharge position GL [LG] 360°
IO position at 315°

Arrangement options viewed on the motor

The inspection opening position is always indicated in degrees of the circular housing. Direction of rotation always clockwise (righthanded) seen from the driven end. This is irrespective of the direction of rotation of the fan.
Arrangement of structural design MAE

Installation and mounting options and corresponding motor types

Fan arrangement

- IMV1
- IMB5
- IMV3

*KSA = drain. The drain, if any, is arranged as shown.

Fan sense of rotation and position options of inspection opening (IO)

- Sense of rotation seen from the driven end clockwise = GR [RD]
- IO position operations viewed on the motor
- Sense of rotation seen from the driven end counterclockwise = GL [LG]

IO position always in clockwise direction seen from the driven end. This is irrespective of the fan’s sense of rotation.
Design characteristics

**Inlet design**
Our optimum aerodynamic flow design includes inlet, inlet cone and nozzle and ensures high efficiency degrees.

1 - inlet  
2 - inlet cone  
3 - nozzle  
4 - impeller  
5 - fan housing  
6 - shaft seal

**Design of fan connections**
The fan is connected to the plant system with flanges according to DIN 24154 R2 edition July 90 or with flat frames as per DIN 24193 R3. In case the fan is connected to duct work with flexible connections with hose clamps, the supplied fan can be of end piece design according to the dimensions of the dimension sheet upon request.

**Sealing of the shaft passage**
The standard design shaft passage is sealed by a belt ring for temperatures up to +80°C. At higher temperatures, the shaft passage is sealed with an asbestos-free flat mechanical seal. This type of sealing is not absolutely tight. Please enquire for superior sealing when fans are subject to special conditions.

1 - housing back plate  
2 - flat seal  
3 - impeller hub  
4 - seal locking plate  
5 - fastening screw
Design characteristics

Fan shaft bearings
Fan shafts for types KX and RG run in two anti-friction bearings. The bearing housings have re-lubricating devices and a grease quantity control. Depending on the type of bearing, bearings are fixed on the fan shaft by conical bearings with clamping sleeves or with cylindrical bearing seats.

- bearing with clamping sleeve (figure 1)
- bearing with cylindrical sleeve (figure 2)
- multiple bearing block with cylindrical bore (figure 3)

1 - self-aligning ball or roller bearing as required
2 - single bearing
3 - grease quantity control
4 - shaft nut
5 - flat lubrication nipple
6 - felt seal
7 - additional seal against dust and dirt

Cooling system
Fans are provided with a heat flinger made of cast aluminium alloy when they handle gas whose temperature exceed 80°C. This heat flinger has heat-dissipating surfaces to reduce the heat radiation to the bearings (fig.4).

1 - fan housing
2 - single bearing housing
3 - fan shaft
4 - shaft seal
5 - heat flinger
6 - guard
Selection of drive motors

The very best branded three-phase standard motors of the protection system IP 55 are used for the drive. The motors are sufficiently dimensioned due to a power consumption that includes a safety margin. If the motors are installed in rooms with a cooling air temperature higher than 40°C, it has to be considered that the output will be reduced. Take values from the manufacturer's motor catalogue.

Protective motor devices

Protective motor devices are used to protect the motors from unacceptable temperature rise, to protect them against damage and to minimize the downtime of electric drives. The efficiency of a motor protection varies according to the applied technical means from most uncomplicated devices that respond often inaccurately only to the coarsest sources of faults, to most expensive and sophisticated equipment which allows to supervise all imaginable possibilities of danger. The inherent high starting current of all radial fans quickly heats up the stator and rotor windings, their temperature will be very high within a few seconds. Therefore it is important for a plant designer to consider the starting time. Starting times from 6 to 10 seconds are referred to as normal starts (the tripping time of the employed protective unit must exceed the starting time of the fan). Starts which take longer are referred to as heavy-duty starts.

Standard value

\[ t_A = 6...10 \text{s} = \text{normal start} \]
\[ t_A \geq 6...10 \text{s} = \text{heavy-duty start} \]

Performance of motor protective devices

The most important function of a motor protective device is to respond before the motor surpasses its maximum permissible excess temperature. However, motor protective devices must not respond if the motor is
- continuously operated with rated power,
- during the allowed run-up time with the allowed start-up current,
- overloaded in warm condition for 2 min with an allowed 1,5-fold nominal current according to DIN VDE 0530.

Explosion protection

Every motor must be protected with a power cut-out switch in order to meet the requirements of protection against explosion in the factory. When selecting the type of switch, care must be taken to ensure that in the event of a short circuit in the motor (i.e. locked rotor) the switch responds within the time \( t_E \) stated on the performance data plate, this must be initiated in accordance with the characteristics in the cold state (20 °C). With reference to the heating-up time \( t_E \) set out in the test regulations, the start-up conditions on motors with the „e“ protection system must be checked with particular care. The generally permitted run-up time for motors with „e“ protection system is as follows:

\[ t_A \leq 1.7 \times t_E \]

For run-up times of \( t_A \) within the \( t_E \)-time range, the protection of current-monitored motors becomes difficult because unnecessary triggering of the overload protector can be effected where start up is repeated, or because the required triggering is not effected despite exceeding the temperature limit of the stator or rotor winding, since in the meantime the overload trigger has cooled more rapidly than the motor because of its smaller thermal time constant. Please contact the motor supplier for further information.
Starting behaviour

Start-up of radial fans

Fans are heavy-starting machines. To start them, the drive motor must overcome the mass moment of inertia of the impeller and when starting against the system resistance it must also overcome the load moment of the fan. Radial fans have a square-law rising load moment (see diagram load torque curve). This can produce unacceptable start-up times and, depending on the individual type of starting (direct or star – delta), run-up problems: the fan possibly does not run up to rated speed. Fans should therefore be started with closed damper where possible (shutter, damper, louvre damper or inlet guide vane). All the motor sizes recommended in our list have been defined accordingly. The run-up times stated in the type selection were calculated for motors whose start-up torque is \( M_s = 2.2 \times \text{rated torque} \). Different start-up torque levels of the individual makes will produce different run-up times.

Load torque curve

Start-up data in design point
1. Load moment
   with open damper
   \( M_L = 9550 \times \left( \frac{P_W}{n_V} \right) \)
2. Load moment
   with closed damper
   \( M_L = 9550 \times \left( \frac{P_W}{n_V} \right) \times 0.5 \)
3. Calculated load moment
   with open damper
   \( M_{Lm} = 9550 \times \left( \frac{P_W}{n_V} \right) \times 0.4 \)
4. Calculated load moment
   with closed damper
   \( M_{Lm} = 9550 \times \left( \frac{P_W}{n_V} \right) \times 0.2 \)

When the motor is ordered and supplied by customer, the following data must be notified to the motor manufacturer before selecting the motor:

1. Fan speed
2. Mass moment of inertia of the impeller
3. Power requirement in nominal point
4. Starting the motor:
   a) with open damper
   b) with closed damper
5. Motor run-up:
   a) in star-delta
   b) direct online
6. Switching frequency or number of start-ups per hour

The size of the motor and type of start-up can only be finally determined by the motor manufacturer after all data has been checked.
Start-up time
Amongst other factors, the start-up time depends on the acceleration torque. The calculated acceleration torque is the difference between the motor torque and the load moment. A precise calculation can only be done by using integral calculus. In practice it is sufficient to ascertain the calculated acceleration torque and thereby to calculate the start-up time.

Calculated acceleration torque
\[ M_{BM} = \text{calculated acceleration torque} \]
\[ M_{Mm} = \text{calculated motor torque} \]
\[ M_{Lm} = \text{calculated load moment} \]
\[ M_A = \text{start-up torque} \]
\[ M_S = \text{pull-up torque} \]
\[ M_K = \text{breakdown torque} \]
\[ M_M = \text{motor torque} \]
\[ M_N = \text{rated torque} \]
\[ M_L = \text{load moment} \]
\[ P_M = \text{rated motor power in kW} \]
\[ P_{Nv} = \text{required shaft power in kW} \]
\[ n_M = \text{motor speed (nominal speed)} \]
\[ n_0 = \text{idling speed} \]

Star-delta start-up
On star-delta start-up, only about 1/3 of the start-up torque is applied by the drive motor in the star circuit. Above a specific start-up speed the load torque of the fan exceeds the starting torque of the motor. The motor will not run up to rated speed. It must therefore be switched to direct online in time during the start-up period. However, this always produces a current peak.

Start-up current
The power supply, the switching devices and the monitoring equipment should essentially be checked by the client with regard to the type of start-up and current peak, ensuring that the dimensions are sufficient. Particular attention should be paid to the start-up current for direct switching. Depending upon the individual rotor category, this is 6 to 8 times the nominal current and must be electrically protected during run-up.

Start-up problems
The motor can only start-up with the fan if sufficient acceleration torque is available throughout the start-up range up to the nominal speed. Subsequent alterations to the absolute course of the motor torque cannot be implemented, thus the correct choice of motor, including the associated switching and monitoring controls, must be made on designing. It is recommended that PTC thermistors protect the motor.

Start-up current too high
Where local mains are too weak, a start-up coupling must be provided or converter technique must be applied on designing the plant. Changes of the fan dimensions should be taken into account.

Use of start-up coupling
Most start-up couplings – hydraulic or mechanical with flyweights – have a torque (friction moment) transferable from the coupling that is equivalent to the square of the RPM speed. For acceleration, the difference between motor and coupling momentums is available. The motor is virtually idling when running-up. The start-up time is lightly higher than the idle-running time. The high start-up current is therefore only of short duration. After this, when the coupling momentum exceeds the load momentum, the coupling’s continuous slip moment accelerates against the load torque of the operating motor. The size of the additional inertia momentum has a significant effect on the run-up time of the machine. The run-up time can vary in the duration.
Operating conditions

Types of installation
Fans can be installed in a technical air system in different ways. In general, fans can be installed in four different types as per DIN 24163 T1.

<table>
<thead>
<tr>
<th>type of installation</th>
<th>description of installation</th>
<th>figure</th>
<th>operation mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>free inlet free discharge</td>
<td></td>
<td>Not permitted for radial fans. Motor will be overloaded.</td>
</tr>
<tr>
<td>B</td>
<td>free inlet discharge ducted</td>
<td></td>
<td>pressure operation</td>
</tr>
<tr>
<td>C</td>
<td>inlet ducted free discharge</td>
<td></td>
<td>vacuum operation</td>
</tr>
<tr>
<td>D</td>
<td>inlet ducted discharge ducted</td>
<td></td>
<td>mixed operation</td>
</tr>
</tbody>
</table>

Installation
In general, fans should be provided with flexible connections. Flexible connections prevent the transmission of structure-borne sound and vibrational forces. They are also supposed to avoid the transmissions of forces from the duct to the fan. At the same time, alignment errors of the duct work are compensated. The flexible connections should always be arranged directly at the fan connection flange except when a damper is installed. If the fan is mounted on anti-vibration mounts, flexible connections must always be provided at inlet and discharge.

Maintenance
The antifriction bearings of the fans of KX and RG designs are lubricated with grease and are designed for a theoretic service life time of at least 40,000 operating hours. The belt drive of the „R“ type is fitted with standard narrow V-belts with a service life time of at least 25,000 operating hours. The main maintenance points include bearings, coupling, belt drive, shaft seals and parts subject to wear and tear such as the impeller. An inspection opening must be provided for checking of the impeller condition. The impeller must be checked for wear and tear or sticking of dust and dirt at regular intervals determined by the degree of wear and tear or soiling level of the handled gas. When carrying out these checks the impeller must be examined in particular for any formation of cracks on the weld seams. Easy and quick accessibility of parts which need to be serviced is important. Appropriate lifting gears and the required space must be provided if necessary for assembly and dismantling. The impeller in particular must be easily accessible.

Monitoring
The type of fan monitoring is determined primarily by its function and its importance within a system. So, depending upon its importance for the system the following factors should be taken into account:

- replacement fan or standby unit
- monitoring devices such as:
  - bearing temperature and bearing condition monitoring
  - speed monitoring
  - vibration monitoring etc.

To ensure high operational safety it is recommended to install a vibration monitoring system in fans that are subject to special load. The system can be set to trigger a preliminary warning or a major alarm or to cause an automatic shutdown even in cases of minor irregularities. Optimum service life and safe operation can only be achieved, however, if proper maintenance is provided and regular checking of parts at risk is carried out. The maintenance and inspection instructions should therefore be precisely observed.
Formulas and units

SI-unit for pressure
One pascal equals the amount of pressure acting constantly on a surface at which a force of 1 N is exerted on 1 m² surface.

\[ 1 \text{ Pa} = 1 \text{ N/m}^2 \]

All pressures in this list are stated in daPa.

\[ 1 \text{ daPa} = 1.02 \text{ mm WC (water column)} \]

Total pressure increase Δpt
The total pressure increase is the total energy from the sum of the static and dynamic pressures.

\[ Δpt = pst + pd \]

Static pressure pst
This is the internal pressure of a gas and it is exerted vertically to the duct wall.

\[ pst_1 ; pst_2 \]

Dynamic pressure pd
This corresponds to the kinetic energy of a flowing gas. This pressure is a function of speed “c” according to the following formula:

\[ pd = \frac{1}{2} c^2 \cdot \rho \]

with \( c \) in m/s

Conversion of pressure units

<table>
<thead>
<tr>
<th>given</th>
<th>required</th>
<th>daPa</th>
<th>Pa N/m²</th>
<th>mbar</th>
<th>bar</th>
<th>mmWS kp/m²</th>
<th>Torr</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 daPa</td>
<td>1</td>
<td>1</td>
<td>10</td>
<td>0.1</td>
<td>10⁻⁴</td>
<td>1.02</td>
<td>7.5 · 10⁻²</td>
</tr>
<tr>
<td>1 Pa</td>
<td>1 N/m²</td>
<td>0.1</td>
<td>1</td>
<td>10⁻²</td>
<td>10⁻⁵</td>
<td>0.102</td>
<td>0.75 · 10⁻²</td>
</tr>
<tr>
<td>1 mbar</td>
<td>10</td>
<td>10²</td>
<td>1</td>
<td>10⁻³</td>
<td>0.102·10²</td>
<td>0.75</td>
<td></td>
</tr>
<tr>
<td>1 bar</td>
<td>10⁴</td>
<td>10⁵</td>
<td>10³</td>
<td>1</td>
<td>0.102·10⁵</td>
<td>750</td>
<td></td>
</tr>
<tr>
<td>1 mmWS</td>
<td>1 kp/m²</td>
<td>0.981</td>
<td>9.81</td>
<td>9.81·10⁻²</td>
<td>9.81·10⁻⁵</td>
<td>1</td>
<td>735 · 10⁻⁴</td>
</tr>
<tr>
<td>1 Torr</td>
<td>13.3 · 10²</td>
<td>13.3 · 10²</td>
<td>1.33</td>
<td>1.33·10⁻³</td>
<td>13.6</td>
<td>1</td>
<td></td>
</tr>
</tbody>
</table>

Volume flow
The volume flow \( \dot{V} \) is the product of the volume and the related time.

\[ \dot{V} \ [m^3/s, m^3/min, m^3/h] \]

The volume flow is always referred to the vacuum operation, i.e. to the static pressure \( pst_1 \), in the inlet and to the inlet temperature \( t_1 \).

\[ \dot{V}_i = \frac{\dot{m}}{p_1} \]

If the temperature changes for operational reasons, the volume flow, however, is maintained.
Formulas and units

**Mass flow \( \dot{m} \)**
The mass flow is the product of the mass and the related time.
\[
\dot{m} = \dot{V} \cdot \rho \quad [\text{kg/s, kg/min, kg/h}]
\]

**Standard cubic metres**
The flow volume in standard cubic meters refers to the physical standard state when the temperature is 0°C and the air pressure 101,325 Pa.
\[
\dot{V}_N = \frac{\dot{V}_N}{\rho_N} = \frac{\dot{V}_N}{\rho_1}
\]
\( \rho_1 \) in kg/m³ = density in the operating state
\( \rho_N \) in kg/m³ = standard density at 0°C and 101,326 Pa

**Total specific supply \( Y_t \)**
The specific total supply is the effective energy difference between the inlet and the discharge referred to the mass.
\[
Y_t = K \cdot \frac{\Delta pt}{\rho_1} \quad [\text{J/kg, daJ/kg}]
\]
\[
Y_t = \frac{\Delta pt}{\rho_m} \quad [\text{J/kg, daJ/kg}]
\]
\( K \) = compression factor
\( \rho_m \) = average density = 1/2 \((\rho_1, \rho_2)\)

**Compression factor \( K \)**
The compression factor \( K \) accounts for the compressibility of the air.
\[
K = \rho_1 \cdot \frac{Y_t}{\Delta pt}
\]
It can be determined with the mean density \( \rho_m \).
\[
K = \frac{\rho_1}{\rho_m}
\]
Thumbrule for the calculation of \( K \)
\[
K = 1 - 0.31 \cdot \frac{\Delta pt}{\text{pa}} \quad ; \quad \frac{\Delta pt}{\text{pa}} \leq 0.1
\]
\[
K = 0.994 - 0.25 \cdot \frac{\Delta pt}{\text{pa}} \quad ; \quad \frac{\Delta pt}{\text{pa}} > 0.1 \leq 0.3
\]
\( \Delta pt \) = static pressure difference
\( \text{pa} \) = absolute pressure
Compression heating in the fan with reference to the nominal point (NP)

The air inside the fan is subject to compression, the temperature rises from the inlet temperature $t_1$ to the discharge temperature $t_2$. The difference between the inlet and discharge temperature is the temperature increase or compression heating $\Delta t$ with $\Delta t = t_2 - t_1$ in °C. The temperature increase can be easily calculated by approximation at a given fan efficiency level with the formula.

$$\Delta t = \frac{\Delta pt}{121 \cdot K}$$

- $\Delta t$ in °C
- $\Delta pt$ in daPa
- $K$ = compression factor (set as 1 in the formula)
- $\eta_i$ = internal fan efficiency

Temperature increase inside the fan with reference to the nominal point and in dependence of the total pressure increase

<table>
<thead>
<tr>
<th>$\eta_i$</th>
<th>$\Delta t$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.7</td>
<td>53°C</td>
</tr>
<tr>
<td>0.25</td>
<td>15.5°C</td>
</tr>
<tr>
<td>0.85</td>
<td></td>
</tr>
</tbody>
</table>

Important remark on fan safety

The fan efficiency deteriorates when the flow is strongly reduced by dampers (deviation from nominal point). Thus, the temperature rises continuously. Temperatures of much more than 50°C can occur on the housing surfaces of high-pressure fans with a total pressure increase of $\Delta pt > 1,000$ daPa. This is to be observed for reasons of the prevention of accidents. The diagram shows a strong reduction of flow ($\eta_i = 0.25$) as a standard value. When the dampers are completely closed (i.e., operation with no flow) the maximum temperature increase is to be assessed.
Formulas and units

Total supply output $P_t$
The total supply output is the product of the mass flow $\dot{m}$ and the total specific supply $Y_t$.

$$P_t = \frac{\dot{m} \cdot Y_t}{100} \quad [\text{KW}]$$

$\dot{m}$ [kg/s] $Y_t$ [daJ/kg]

Overall efficiency $\eta_{tW}$
Overall efficiency is to be understood as the total supply $P_t$ to shaft power $P_W$ ratio without V-belt drive losses.

$$\eta_{tW} = \frac{P_t}{P_W} \quad P_t ; P_W \quad [\text{KW}]$$

Required shaft power $P_W$
The required shaft power $P_W$ is the power taken up at the fan shaft including the corresponding mechanical losses such as bearing friction and coupling losses.

$$P_W = \frac{\dot{m} \cdot Y_t}{100 \cdot \eta_{tW}} \quad [\text{KW}]$$

$\dot{m}$ [kg/s] $Y_t$ [daJ/kg]

$$P_W = \frac{V_t \cdot \Delta p_t}{6000 \cdot \eta_{tW}} \cdot K \quad [\text{KW}]$$

$V_t$ [m³/min] $\Delta p_t$ [daPa]

$\eta_{tW}$ = overall efficiency of the fan referred to the fan shaft.
For total pressures up to $\Delta p_t = 355$ daPa the compression factor can be neglected.

$$P_W = \frac{V_t \cdot \Delta p_t}{6000 \cdot \eta_{tW}} \quad [\text{KW}]$$

for $\Delta p_t \leq 355$ daPa

Efficiency $\eta_{tW}$ (without compression factor)

$$\eta_{tW} = \frac{\Delta p_t \cdot V_t}{6000 \cdot P_W} \quad \Delta p_t \text{ [daPa]}$$

$V_t$ [m³/min] $P_W$ [KW]

Efficiency $\eta_{tW}$ (with compression factor)

$$\eta_{tW} = \eta_{tWV} \cdot K$$
Formulas and units

Effect of density
In case of ideal gases the density can be determined from the general gas equation.

\[ p = \frac{p}{R \cdot T} \quad [\text{kg/m}^3] \]

\( p \) = pressure \([\text{Pa}]\)
\( T \) = thermodynamic temperature \([\text{K}]\)
\( R \) = gas constant \([\text{J/kgK}]\)

At different air density levels the pressure and the shaft power changes in proportion to the density. By contrast, the flow volume is kept constant.

\[ \frac{\Delta p_I}{\Delta p_{II}} = \frac{\frac{p_I}{\rho_I}}{\frac{p_{II}}{\rho_{II}}} ; \quad \frac{P_{II}}{P_{II}} = \frac{\frac{p_I}{\rho_I}}{\frac{p_{II}}{\rho_{II}}} \]

Influence of air pressure
Dependent on the altitude of the installation site, the absolute air pressure and consequently the corresponding air density changes.

\[ p_a = p_0 \cdot \left[1 - \frac{6.5 \cdot h}{273 + t}\right]^{5.256} \quad [\text{Pa}] \]

\( p_a \) = absolute air pressure in „h“ m altitude \([\text{Pa}]\)
\( p_0 \) = reference air pressure in „0“ m altitude \([\text{Pa}]\)
\( h \) = altitude of site \([\text{km}]\)
\( t \) = temperature \([\text{°C}]\)

\[ \rho_a = \frac{p_a}{R \cdot T} \quad [\text{kg/m}^3] \]

Speed change
The flow volume alters in proportion to the speed.

\[ \frac{V_{II}}{V_{II}} = \frac{n_{II}}{n_{II}} \]

The pressure alters in proportion to the square of the speed ratio.

\[ \frac{\Delta p_{II}}{\Delta p_{II}} = \left(\frac{n_{II}}{n_{II}}\right)^2 \]

The shaft power changes in proportion to the speed ratio to the third power.

\[ \frac{P_{II}}{P_{II}} = \left(\frac{n_{II}}{n_{II}}\right)^3 \]

Index:
I = initial condition
II = altered condition
Fan performance curve

General information
The fan performance curve always indicates the pressure for the corresponding volume flow. The indicated pressure is always the total pressure increase $\Delta p_t$. For finding the static pressure $p_s$, subtract the dynamic pressure $p_d$ from $\Delta p_t$. The graph of the performance curve depends on various geometric and ventilation parameters. Therefore, the performance curve type is indicated in the type selection for each type of fan. The graph of the performance curve should be taken from the respective performance curve (see sheets TA 46 to TA 52).

Nominal point NP
The ratings indicated in our type selection are within the optimal range of efficiency and are designated as nominal point NP.

Plant characteristic curve
There is a resistance to the fan in every plant. For most ventilation plants the resistance runs in the form of a parabola. This characteristic curve has to be carefully calculated by the client.

Operating point BP
As the fan can operate at any point of its performance curve depending on the plant resistance, the actual working point in the plant is called the operating point, OP.

Interaction of plant and fan
The interface point of the fan performance curve and the plant characteristic curve is the actual operating point OP. In the best case, it is close to the nominal point NP. Should the plant resistance be lower than calculated ($\Delta p_{tI}$), there will be a higher volume flow ($V_{1i}$) at the operating point BP. As a result, the mounted motor will be overloaded and damaged. Especially for fans with a constant increase in power consumption (KL type 1, 2 and 3), the power consumption increases even when the nominal point is only slightly exceeded.

NP = nominal point = list data
BP = operation point = working point in the plant

Reserve power
For this reason it is recommended to apply a motor with sufficient reserve. Experience shows that it is advisable to choose a driving power that is about 15-30% higher than the required shaft power.

$P_M = P_W + 15\% \text{ to } 30\%$
Pressure course at different operation modes

Pressure operation

\[ \dot{V}_2 = \dot{V}_1 \]
\[ \rho_0 = \rho_1 \]
\[ \Delta p_{t2} = \Delta p_{t1} - \rho_2 \]
\[ \Delta p_{t1} = \Delta p_{t2} - \rho_2 \]

\[ \rho_0 = \text{reference density} \]
\[ \rho_0 = \text{reference air pressure} \]
\[ \Delta p_{t2} = \text{total pressure increase - discharge} \]
\[ \rho_{st2} = \text{static pressure - discharge} \]
\[ \rho_{d2} = \text{dynamic pressure - discharge} \]

Vacuum operation

\[ \dot{V}_0 = \dot{V}_{1; \rho_0} \]
\[ \dot{V}_1 = \dot{V}_{0; \rho_0} \]
\[ \rho_0 = \rho_1 \]
\[ \Delta p_{t1} = \Delta p_{t2} - \rho_2 \]
\[ \rho_{pt1} = \Delta p_{t1} - \rho_2 + \rho_1 \]

\[ \Delta p_{t1} = \text{total pressure increase - inlet} \]
\[ \rho_{pt1} = \text{static pressure – inlet (observe preceding sign for } \rho_{pt1} \text{)} \]
\[ \rho_{d1} = \text{dynamic pressure - inlet} \]
\[ \rho_{d2} = \text{dynamic pressure – discharge} \]

In case of operation at inlet the flow to be handled is always to be considered and indicated. The volume flow \( \dot{V}_0 \) at the intake or vacuum point is always smaller than the volume flow \( \dot{V}_1 \) at the fan inlet. To facilitate conversion from \( \dot{V}_0 \) to \( \dot{V}_1 \), the corresponding conversion factors \( f_{\rho0} \) are included in the type selection sheets (see TA 2).

\[ \dot{V}_1 \cdot \rho_1 = \dot{V}_0 \cdot \rho_0 \]
\[ \dot{V}_0 = \dot{V}_1 \cdot f_{\rho0} \]
\[ f_{\rho0} = \rho_1 / \rho_0 \text{ with } \rho_0 = 1.205 \text{ kg/m}^3 \]

Mixed operation

\[ \Delta p_t = \rho_{st2} + \rho_{pt1} + \rho_{d2} + \rho_{d1} \]
Performance curve courses at different operation modes

Pressure operation

\[ \Delta p_{t2} = p_{st2} + p_{d2} \]
\[ p_{st2} = \Delta p_{t2} - p_{d2} \]

\[ \rho_2 = \frac{(p_0 + p_{st2}) \cdot 10}{R \cdot T_2} \]
\[ c_2 = \frac{\dot{V}_2}{60 \cdot A_2} \]
\[ p_{d2} = \frac{\rho_2}{20} \cdot c_2^2 \]

\[ A_2 = \text{area of discharge} \]

Vacuum operation

\[ \Delta p_{t1} = p_{st1} + p_{d2} - p_{d1} \]
\[ p_{st1} = \Delta p_{t1} - p_{d2} + p_{d1} \]

\[ \rho_1 = \frac{(p_0 - p_{st1}) \cdot 10}{R \cdot T_1} \]
\[ c_1 = \frac{\dot{V}_1}{60 \cdot A_1} \]
\[ p_{d1} = \frac{\rho_1}{20} \cdot c_1^2 \]
\[ \Delta p_{d} = p_{d2} - p_{d1} \]
\[ \text{if } A_1 = A_2 \text{ then } \Delta p_{d} = 0 \]
\[ \Delta p_{t1} = p_{st1} \]

\[ A_1 = \text{area of inlet} \]

Mixed operation

\[ \Delta p_{t} = p_{st2} + p_{st1} + p_{d2} - p_{d1} \]

\[ \Delta p_{d} = p_{d2} - p_{d1} \]
\[ \text{if } A_1 = A_2 \text{ then } \Delta p_{d} = 0 \]

\[ \Delta p_{t} = p_{st2} + p_{st1} \]

Change of density

The change of density from discharge to inlet operation is noticeable in case of a total pressure increase from \( \Delta p_{t2} = 250 \text{ daPa} \) onwards. The type selection indicates the respective total pressure increases for inlet operation \( \Delta p_{t1} \) and for discharge operation \( \Delta p_{t2} \) (see PB 25).
Conversion from pressure to vacuum operation

All fan design data (calculation data) refer to pressure operation based on an inlet temperature of 20°C in the fan inlet, an air pressure (atmospheric pressure) of $p_0 = 10132.5$ daPa and a density of $\rho_0 = 1.205 \text{kg/m}^3$.

For the calculation of the density $\rho_1$ and $\rho_2$, the static pressure is equated with the total pressure increase ($\Delta pt = pst$) and provided that inlet and pressure openings have the same diameter.

Basic formula for the pressure conversion

In the conversion from pressure to vacuum operation and vice versa, the pressures behave like the densities in pressure and vacuum operation under atmospheric condition.

$$
\frac{p_1}{p_0} = \frac{p_2}{\rho_2} = \frac{\Delta pt_1}{\Delta pt_2} \quad \text{with} \quad \rho_2 = \frac{\left(\rho_0 + \Delta pt_2\right) \cdot 10}{287 \cdot (273 + 20)}
$$

$$
\frac{\Delta pt_2}{\Delta pt_1} = \frac{p_2}{p_1} \quad \text{with} \quad \rho_1 = \frac{\left(\rho_0 - \Delta pt_1\right) \cdot 10}{287 \cdot (273 + 20)}
$$

$\rho_0 = 1.205 \text{ kg/m}^3$

**Formula symbols**

- $\Delta pt = $ total pressure increase
- $pst = $ static pressure
- $pd = $ dynamic pressure
- $V_t = $ volume flow in m$^3$/min
- $A = $ surface area in m$^2$
- $NP = $ fan nominal point
- $\rho_0 = 1.205 \text{ kg/m}^3 = $ reference density
- $\rho = $ density in kg/m$^3$
- $T = $ thermo-dynamic temperature in K
- $t = $ temperature in °C
- $R = $ gas constant for air = 287 J/kg · K

**Index:**

- 1 = inlet
- 2 = discharge
- 0 = reference figures with
  - $p_0 = 10132.5 \text{ daPa atmospheric pressure}$
  - $t_0 = 20\degree\text{C inlet temperature}$
  - $\rho_0 = 1.205 \text{ kg/m}^3 \text{ inlet density}$
  - $p = \text{ absolute pressure in daPa}$

**Level:**

- 0: reference pressure = atmospheric pressure
- 2: operation at discharge (positive pressure)
- 1: operation at inlet (negative pressure)
Operational behaviour

Installation loss
Both fan and plant are part of an interacting complex aerodynamic system. After the installation in a plant, there will consequently be a deviation from the standard performance curve depending on the kind of the inlet and discharge conditions. The deviations will be indicated as installation loss.

Deviation from the standard performance curve

\[ \Delta p_t = \Delta p_{tNP} \]

In order to keep the installation loss as small as possible the following frequent malfunction sources should be avoided:
- bend directly up- or downstream of the fan
- dampers directly upstream of the inlet
- flexible connections with constricted cross-section
- inlet duct too short
- improper transition pieces
- pressure loss caused by suction box
- inlet whirl by components at inlet
- leakage loss in ducts and plant components
- stalls caused by plant
- dynamic pressure at discharge (diffuser)

Standard performance curve
According to DIN 24163 the standard performance curve of a fan is the interrelation between the fan pressure increase \( \Delta p_t \) and the handled flow rate \( V_t \) metered on the test bench under accurate test conditions. To complete the operating data, the power consumption \( P_W \) is also indicated. All data given in the technical literature is principally based on this standard performance curve. The performance of the standard curve depends on various geometrical and flow parameters. The design or calculation point of the performance curve is called the nominal point (NP). It is within the optimum range of efficiency. All ventilation data refers to the following reference figures:

- air pressure \( p_0 = p_a = 10132.5 \text{ daPa} \)
- inlet temperature \( t_0 = t_1 = 20^\circ\text{C} \)
- inlet density \( \rho_0 = \rho_1 = 1.205 \text{ kg/m}^3 \)
Tolerances

Basic criteria
Certain deviations from the agreed operational values are permissible due to unavoidable design, calculation and production tolerances (called "as-built tolerances"). The permissible deviations depend on the class of accuracy of the fan. The selection of the accuracy class for the individual fan depends on different criteria. It might be necessary to adapt the accuracy class to the ambient or operating conditions. Uncertainties concerning the defining of the operational values due to special installation conditions (e.g. disturbances at the inlet and discharge) are not included in the as-built tolerances and must be considered separately. For further information see our brochure „Fans in practice“.

Tolerances depending on the class of accuracy

<table>
<thead>
<tr>
<th>class of accuracy according to DIN 24166</th>
<th>0</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>volume flow ( \dot{V}_l )</td>
<td>±1%</td>
<td>±2.5%</td>
<td>±5%</td>
<td>±10%</td>
</tr>
<tr>
<td>total pressure increase ( \Delta p_t )</td>
<td>±1%</td>
<td>±2.5%</td>
<td>±5%</td>
<td>±10%</td>
</tr>
<tr>
<td>shaft power ( P_{W_1} )</td>
<td>+2%</td>
<td>+3%</td>
<td>+8%</td>
<td>+16%</td>
</tr>
<tr>
<td>efficiency ( \eta_{tw} )</td>
<td>-1%</td>
<td>-2%</td>
<td>-5%</td>
<td>-</td>
</tr>
<tr>
<td>sound values ( L_{W_1}, L_p )</td>
<td>+3dB</td>
<td>+3dB</td>
<td>+4dB</td>
<td>+6dB</td>
</tr>
</tbody>
</table>

In case no special specifications have been agreed upon, the following accuracy classes apply:

- shaft power \( P_{W_1} > 50kW \) class 1
- shaft power \( P_{W_1} < 50kW \) class 2

Special design fans (e.g. unshrouded impellers, rubberlined or coated impellers, impellers with strongly curved blades) are class 3. Class 2 applies to slightly modified impellers flow conditions.

Operational condition
The tolerances apply only to design point or nominal point (NP) of the fan. It has been specified with regard to the speed, volume flow, pressure, density and handled gas.

As-built tolerances
The allowed deviations in the dimension sheets comply with ISO 2768-mK and EN ISO 13920-A.

<table>
<thead>
<tr>
<th>nominal range (mm)</th>
<th>above 6 to 30</th>
<th>above 30 to 120</th>
<th>above 120 to 315</th>
<th>above 315 to 1000</th>
<th>above 1000 to 2000</th>
<th>above 2000 to 4000</th>
<th>above 4000 to 8000</th>
</tr>
</thead>
<tbody>
<tr>
<td>tolerance (mm)</td>
<td>+1</td>
<td>+1.5</td>
<td>+2</td>
<td>+3</td>
<td>+4</td>
<td>+5</td>
<td>+8</td>
</tr>
</tbody>
</table>
Sound behaviour

For design and construction of ventilation plants, compliance with the given noise limits is necessary to protect the neighbourhood from sound irritation. The fan in particular is one of the most critical sound sources within the entire plant which should be paid particular attention to.

In order to maintain and prove the agreed sound specifications, it is necessary to measure noise according to the given standard regulations.

For sound measurements at fans, standardised regulations apply as described in DIN 45635, sheet 1 „Machine noise measurement“ and DIN 45635 part 38 „Fan noise measurement“.

The standard describes the precondition for determination of the sound radiated directly from the fan into the environment (sound emission), according to standardised methods, so that the results can be compared. The measuring method described in the standard is only valid for free sound radiation i.e. in a reflection-free environment.

In practice, however, there are generally no optimal terms. Noise values measured under operational conditions differ more or less from the values measured in reflection-free space. The individual operating conditions and type of installation in combination with the environmental influences normally lead to considerable increases in the noise levels. The provision of a guarantee to the end-user is subject to the consideration of plant-specific additions and acoustic calculations.

\[
\text{fan noise level in the system} = \text{REITZ fan noise level specification} + \text{addition}\text{(adjustment)}
\]

*addition from 3 to 9 db, dB(A) are quite realistic

The plant designer or acoustic engineer has to ascertain and calculate the additions. The empiricals for the addition depend on the number of parameters which can be influenced.

Influence of noises under operating conditions

In order to transfer measurements for the fan taken in optimum conditions to working conditions, it is indispensable to observe and take into account the following sources of interference:

- noise of drive motors
- background noise generated by other machines
- level increase by room influence (reflection)
- level increase by deviation from nominal point (fan is deviating from order values when operating in the plant)
- level increase caused by dampers (inlet guide vanes, valves, shutters, etc.)
- level increase caused by flexible connections (they represent areas of “noise leaks“ in the system)
- level increase caused by plant components as for instance ducts, bends, baffles, suction boxes, changes of cross section, transition pieces etc.
- level increase caused by stalls in the plant

In principle, the sources of interference produced by the plant itself as well as interference caused by set-up (locality) are to be calculated and determined by the designer of the plant. Please refer to our brochure entitled „Practical sound design“. This detailed information on sound is intended to avoid design and planning faults.