Constant design air flow industrial ventilation systems with regenerative dust filters: Economic comparison of fan speed-controlled, air damper controlled and uncontrolled operation

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A B S T R A C T
Variable speed drives for fans in small and medium-size ventilation systems with a constant design air flow, including a regenerative filter for dust separation, are evaluated in this paper. In the past, variable speed drives were considered for systems with varying air flow. It was found that this is also an appropriate measure to reduce the operating costs of such systems. With the chosen approach the calculated payback period based on 24 h operation is between 0.7 and 1.7 years, depending on the size of the system. For systems in a two-shift operation the payback periods are still in the range of 1.4–3.4 years. The profitability of the installation of fan speed-control based on 12,000 service hours is in the range of 80–150%. A new operation mode, in which the filter cleaning is triggered at a fixed pressure drop of the filter cake, would further reduce the payback period and improve profitability. Systems with automatic damper flow control are much less economical. However, for the assumed conditions such systems are still better than uncontrolled systems. The installation of speed-control for fans in ventilation systems with a constant design air flow is therefore highly recommended for an economical design.

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1. Introduction

Most of the power consumption of ventilation systems is caused by the motor of the suction fan which is required to provide the necessary draught and to overcome the pressure drop of the system caused by ducts, dampers, dust filters and other equipment. Therefore, measures for the reduction of the operating costs of such systems often aim to reduce the pressure drop of the system [1]. Another energy-saving strategy in ventilation systems is the use of highly efficient electric motors [2,3]. In many ventilation systems the required air flow is not constant. To save energy the air flow can be reduced to a level that still delivers the required air quality (ventilation on demand). Therefore, the ventilation air volume is controlled by the use of variable speed drives for the fans to match load requirements [2,4,5]. Depending on the variation of the ventilation demand, electrical energy savings by the use of variable speed drives are typically in the range of 25–50% [6]. The other operating alternatives are operation at the maximum capacity of the fan or flow control with a control damper increasing the pressure loss in the suction line. However, the power consumption of the fan motor is higher with both alternatives [7,8].

Numerous studies deal with the energy optimization of heating, ventilation and air-conditioning systems (HVAC) for buildings. In these systems the main purpose of ventilation is the control of the room atmosphere with respect to temperature and carbon dioxide concentration by replacing the air with fresh air. Ventilation on demand by the use of variable speed drives for the ventilation fans was investigated for various kinds of buildings such as detached family houses [9], schools [10–12], university buildings [13] and office buildings [14,15]. This operation enabled a reduction in power consumption of the fan in the range of 35–50% [9,13]. The reduction in the total energy consumption of the fan and the heating were reported to be in the same range [10,16].

Papers on the use of variable speed drives for fans in industrial applications concentrate on two areas: cement kiln fans [17–22] and mine ventilation fans [23–25]. In mine ventilation the control of the atmosphere is essential, whereas cement kiln fans are a part of the kiln off-gas de-dusting system. For cement kiln fans electrical energy savings of 25–40% are reported for speed-controlled operation [17,19]. The reported electrical energy savings in mine ventilation are up to 50% or 60% [25]. Other industrial applications of variable speed drive fans are described for cooler fans [26], fans

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### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ΔPₚ</td>
<td>Pressure drop of the air collection system, Pa</td>
</tr>
<tr>
<td>ΔPₚₑ₉</td>
<td>Pressure drop of the filter medium, Pa</td>
</tr>
<tr>
<td>ΔPₚₑ₆</td>
<td>Pressure drop of the filter cake, Pa</td>
</tr>
<tr>
<td>Pₛₑ</td>
<td>Shaft power of the fan, W</td>
</tr>
<tr>
<td>Pₑₑ₉</td>
<td>Electric power demand of the fan, W</td>
</tr>
<tr>
<td>Pₑₑ₆</td>
<td>Rated power of the fan motor, kW</td>
</tr>
<tr>
<td>τₑₑ₉</td>
<td>Efficiency of the fan</td>
</tr>
<tr>
<td>τₑₑ₆</td>
<td>Efficiency of the electric drive</td>
</tr>
<tr>
<td>Vₑₑ₉</td>
<td>Volumetric flow rate, m³/s (at fan inlet conditions)</td>
</tr>
<tr>
<td>ΔCₙᵢₙₚ</td>
<td>Additional investment costs, €</td>
</tr>
<tr>
<td>ΔPₑₑ₉</td>
<td>Difference in static pressure, Pa</td>
</tr>
<tr>
<td>ρₐₑₑ₉</td>
<td>Ambient pressure, Pa</td>
</tr>
<tr>
<td>ρₛₑ₉</td>
<td>Static pressure, Pa</td>
</tr>
<tr>
<td>nₑₑ₆</td>
<td>Polytropic exponent</td>
</tr>
<tr>
<td>kₑₑ₆</td>
<td>Isentropic exponent</td>
</tr>
<tr>
<td>fₑₑ₆</td>
<td>Factor according VDI 2044</td>
</tr>
<tr>
<td>kₙₑₑ₉</td>
<td>Coefficients in fan volumetric flow calculation</td>
</tr>
<tr>
<td>kₑₑ₉ₑₑ₉</td>
<td>Coefficients in fan efficiency calculation</td>
</tr>
<tr>
<td>A–D</td>
<td>Coefficients in motor efficiency calculation according ÖVE/ÖNORM EN 60034–30</td>
</tr>
<tr>
<td>Rₑₑ₉</td>
<td>Resistance of the system</td>
</tr>
</tbody>
</table>

### Indices

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iₑₑ₆</td>
<td>At fan inlet</td>
</tr>
<tr>
<td>Mₑₑ₆</td>
<td>At design point of the fan (maximum pressure drop)</td>
</tr>
<tr>
<td>Cₑₑ₆</td>
<td>Ventilation system with volumetric air flow controlled by a speed-controlled fan</td>
</tr>
<tr>
<td>Uₑₑ₆</td>
<td>Ventilation system with uncontrolled volumetric air flow</td>
</tr>
</tbody>
</table>

in a boiler house [27] and for laboratory exhaust fans [28]. The reported electrical energy savings are about 50% [28]. The payback period of the additional investment cost for the installation of a frequency converter to enable speed-controlled operation depends on the variation of the required ventilation air flow and the electricity cost per unit. It is generally less than two years [26,27,29,30].

Common to all the above-mentioned applications of variable speed drives for ventilation fans is the variable demand for the ventilation air volume flow. However, in many ventilation systems for processing units e.g. for gas, laser and plasma-cutting, blast machines, grinding, brushing and polishing operations in metalworking or mechanical sand processing in foundries, the required ventilation air flow is constant. However, in these systems the operating conditions for the fan are not constant because the resistance of the installed filter for de-dusting increases gradually due to the separated dust, thus reducing the air flow. The total pressure drop of such a system is the sum of the pressure drop of the air collection system and the pressure drop of the filter, which results from the pressure drop of the filter medium and the pressure drop of the filter cake formed by the separated dust. The pressure drop of the filter medium (residual pressure drop) increases slightly when the filter medium is in use because some dust particles in or on the filter are not removed during the regenerative cleaning process. The pressure drop of the filter cake is zero for the cleaned filter medium and increases during the filtration cycle. The rate of the increase depends on the dust concentration, the dust characteristics (particle size, shape of the particles) and on the specific air flow rate (air to cloth ratio). At the end of each filtration cycle the filter is regenerated by a compressed air pulse detaching the filter cake. After regeneration the pressure drop of the filter cake is again zero. Thus the pressure drop of the filter system is not constant [31,32]. Fig. 1 shows the course of the pressure drop which is usually found in small and medium-sized fabric filters. During each filtration cycle the pressure drop increases from the residual pressure drop of the filter medium to the chosen maximum pressure drop, when the regenerative cleaning is started.

In large filters the cleaning is spread more evenly over time because only a part of the large filter area is cleaned at once. In the cleaned part of the filter area the resistance is reduced. Hence, the pressure drop has to be the same for the whole filter area, the air flow through the cleaned filter area increases, and the flow in the un-cleaned area decreases until the pressure drops are equal in both areas. Thus, the reduction in the pressure drop is smaller in this case.

The aim of this study is to investigate the economic feasibility of the use of frequency converters for speed-controlled operation of the fan and air damper control in small and medium size ventilation systems with a constant design air flow. The evaluation is made on the premise that a new ventilation system is built. The basic design of the ventilation system investigated does not have flow control. It consists of an air collection system, a regenerative fabric filter for dust separation and a centrifugal fan. The additional investment costs for both controlled systems and the energy savings of controlled operation are estimated for both systems. From these results the payback periods for the required additional investments are calculated.

### 2. Ventilation system design

Ventilation systems are usually designed for operation at the maximum pressure drop of the system which is reached just before regeneration of the filter medium is triggered (shown in Fig. 2 as point M). When the filter medium is clean, the pressure drop is smaller. In an uncontrolled system the starting point of each filtration cycle also has to be on the performance curve of the fan. Therefore, the air flow is higher. The corresponding operation point is shown in Fig. 2 as point U. During the build-up of the filter cake the operation point moves gradually from U to M, the regeneration of the filter medium brings the operation point back to U. In the course of the time of use the operation point at cleaned filter medium U is shifted slowly in the direction of M. This is caused by the limited efficiency of the filter regeneration. The efficiency of a fan is not constant either. Usually, the fan is selected to have the highest efficiency at the design point. The more the operation point differs from the design point, the lower the efficiency of the fan. Depending on the gradient of the performance curve and the fan efficiency, it often happens that the power consumption is higher for operation points with a reduced pressure drop.

With the installation of a control damper in the suction line the fan can always be operated at the design operation point (M) because the controlled pressure drop of the damper compensates for the variable pressure drop of the filter cake. In this case the power consumption is constant.

The installation of a frequency converter in the power supply line of the fan motor enables constant air flow operation by speed control. The fan performance curve is shifted towards less capacity (flow and pressure) when the speed is reduced. In this case the operation point with a clean filter medium would be C. In the course of the filtration cycle it moves gradually from C to M as a filter cake is formed by the separated dust and is shifted back to C when the filter is regenerated. With increasing time of use the starting point of the filtration cycle C moves slowly in the direction of M. The efficiency of the fan is also lower at C, but due to the remarkable reduction in pressure drop the fan’s power consumption is much lower.
3. Calculation

3.1. System designs considered

The ventilation systems investigated consist of an air collection system with fixed settings, a fabric filter which is regenerated when a maximum pressure drop is reached and a centrifugal suction fan supplying the required draught. The following designs for air flow control are considered:

- uncontrolled air flow (UC),
- constant air flow controlled by a control damper (CD),
- constant air flow controlled by a speed-controlled fan (SCF).

The flow sheets for the different system designs are shown in Fig. 3. For the system with uncontrolled air flow (Fig. 3a) a motor soft starter and a manual damper upstream of the fan are considered. For the system with constant air flow controlled by a damper (Fig. 3b) an actuator for the damper and flow measurement for the volumetric air flow are additionally required. When the air flow is controlled by a speed-controlled fan (Fig. 3c) a frequency converter replaces the soft starter and shielded cables are required. A flow measurement device is also needed in this system.

3.2. Power consumption of fans

The calculation of the power consumption was programmed in Visual Studio. Assuming equal inlet and outlet air velocity the fan shaft power can be calculated by Eq. (1) as follows:

\[
P_{sh} = \frac{1}{n_{fan}} \cdot \dot{V}_1 \cdot \frac{n}{n-1} \cdot \left[ \left( 1 + \frac{\Delta p_{st}}{p_{st,1}} \right)^{\frac{n-1}{n}} - 1 \right]
\]
where the index 1 relates to the inlet conditions of the fan. According to VDI 2044 [33] in case of fans with a relative pressure increase of Δp_{st}/p_{st,1} ≤ 0.1 and equal air velocity at the inlet and the outlet of the fan, the fan shaft power can be calculated by a simplified equation Eq. (2):

\[ P_{sh} = \frac{\dot{V}_1 \cdot \Delta p_{st}}{\eta_{fan}} \cdot f \]  

For fan efficiencies > 0.5, the isentropic exponent (κ) can be used instead of the polytropic exponent (n). Then, for air (κ = 1.4), the factor f can be calculated by Eq. (3):

\[ f = 1 - 0.36 \frac{\Delta p_{st}}{p_{st,1}} \]  

The electric power demand is given by Eq. (4):

\[ P_{el} = \frac{P_{sh}}{\eta_{drive}} \]  

The required pressure increase of the fan results from the sum of the pressure drops of the air collecting system (Δp_{S}), of the filter medium (Δp_{FM}) and the filter cake (Δp_{FC}). It can be written as shown in Eq. (5):

\[ \Delta p_{st} = \Delta p_{S} + \Delta p_{FM} + \Delta p_{FC} \]  

In typical ventilation systems the fan is situated downstream of the filter close to the air discharge. Therefore, the static pressure at the fan inlet can be approximated by Eq. (6):

\[ p_{st,1} = p_A - \Delta p_{st} \]  

where \( p_A \) is the ambient pressure. Thus, the electrical power demand of the fan can be calculated by Eq. (7):

\[ P_{el} = \frac{\dot{V}_1 \cdot \Delta p_{st}}{\eta_{fan} \cdot \eta_{drive}} \cdot \left( 1 - 0.36 \cdot \frac{\Delta p_{st}}{p_A - \Delta p_{st}} \right) \]  

### 3.3. Performance curves of fans

For the calculation of the performance curves of the centrifugal fans the computer program RKVent® of Reitz Ventilatoren GmbH & Co. KG was used. With this program different types of fans with a power range of up to 1000 kW can be designed. The use of the program is free of charge and it can be downloaded from the company’s website [34]. For each plant size the fan performance and efficiency curves were calculated for optimum fan efficiency at the design operation point. The resulting curves were then digitalized by using the program DataLab® and expressed by polynomial functions Eqs. (8) and (9) which were found through linear regression [35]. The specific constants k0–k3 and h0–h3 for each fan size were included in the calculation model:

\[ \dot{V}_1 = k_0 + k_1 \cdot \Delta p_{st} + k_2 \cdot \Delta p_{st}^2 + k_3 \cdot \Delta p_{st}^3 \]  \hspace{1cm} (8)

\[ \eta_{fan} = h_0 + h_1 \cdot \dot{V}_1 + h_2 \cdot \dot{V}_1^2 + h_3 \cdot \dot{V}_1^3 \]  \hspace{1cm} (9)

### 3.4. Efficiency of motors

The efficiency of a grid-connected motor operated close to its design point can be calculated by the following approximation equation Eq. (10), published in the standard ÖVE/ÖNORM EN 60034-30 [36]:

\[ \eta_{motor} = A + B \cdot \left[ \log_{10}(P_N) \right]^2 + C \cdot \left[ \log_{10}(P_N) \right] + D \]  \hspace{1cm} (10)

The coefficients A–D depend on the nominal power of the motor \( P_N \) (in kW), the efficiency class and number of pole pairs. The coefficient can also be found in this standard.

The efficiency of a drive system consisting of a motor and a frequency converter (FC) can also be taken as a constant. The efficiency of a grid-connected motor of the same size [37,38].

### 3.5. Calculation models for the fan power consumption

In the calculation of the ventilation system with constant air flow controlled by a damper the pressure increase of the fan (Δp_{st}) is equal to the maximum pressure drop (Δp_{PM}). Due to the constant pressure increase also the volumetric air flow is constant.

For a ventilation system design with a speed-controlled fan operated at constant flow the pressure drop starts in each filtration cycle at Δp_{FC}(t) which is the sum of the pressure drop of the air collection system and the residual pressure drop of the filter medium (Eq. (11)):

\[ \Delta p_{FC}(t) = (\Delta p_{S} + \Delta p_{PM}(t)) \]  \hspace{1cm} (11)

At the end of a filtration cycle the maximum pressure drop of the system is reached. Assuming, on average, a linear increase in the pressure drop, the sawtooth-shaped curve of the actual pressure drop was replaced in the calculation model by a curve resulting from the average of both (Eq. (12)):

\[ \Delta p_{st}(t) = \frac{\Delta p_{PM} + \Delta p_{FC}(t)}{2} \]  \hspace{1cm} (12)

Measured functions of the pressure drop are slightly concave [32]. Therefore, the above assumption slightly over-estimates the pressure drop. The development of the residual pressure drop of the filter medium (Δp_{PM}(t)) was approximated by two functions based on measured results of the residual pressure drop at a filter.
unit in the ventilation system for a rolling mill that was operated at a constant air flow of 25,000 Nm\(^3\)/h and an average inlet dust concentration of about 7 mg/Nm\(^3\) [35]. For the first 200 operation hours the linear function \(\Delta P_{PM}(t) = 1.16 \cdot t + 250\) was used. For the time beyond 200 h the function \(\Delta P_{PM}(t) = 322 \cdot \ln(t) - 1225\) delivers a good approximation of the measured results. In both functions the unit of the operation time \(t\) is hours.

For the uncontrolled system the starting point of the filtration cycle is the intersection of the fan performance curve and the characteristic curve of the system. The latter curve is defined by the resistance of the system \(R\) which depends on the actual status of the filter material. Thus, the pressure drop \(\Delta p_U(t)\) and the flow \(V_U\) can be calculated by solving the system of the equations Eqs. (13) and (8) numerically:

\[
R(t) = \frac{\Delta p_C(t)}{V_M^2} = \frac{\Delta p_U(t)}{V_U^2(t)}
\]  

(13)

For uncontrolled operation a linear increase in the pressure drop was assumed. In the calculation model the sawtooth-shaped curve of the actual pressure drop again was replaced by a curve resulting from the average pressure drop. The corresponding air flow was calculated by Eq. (8).

For the speed-controlled fan system (SCF) an additional operation mode (SCF-CPD) was investigated. In this mode the end of a filtration cycle is reached and filter regeneration is started when the pressure drop has increased by a certain value \(\Delta p_{FC}\) within this filtration cycle. In this mode the average pressure drop of the system is lower and the filter cleaning is more frequent, which increases the consumption of compressed air.

In the ventilation system with the constant operation point also the efficiency of the fan \(\eta_{fan,M}\) is constant. It can be calculated using Eq. (9). For the system with the speed-controlled fan and for the uncontrolled system, the average of the fan efficiency at the start and the end of the filtration cycle was used as fan efficiency in the calculation:

\[
\bar{\eta}_{fan}(t) = \frac{\eta_{fan,M} + \eta_{fan}(t)}{2}
\]  

(14)

As the efficiency of a centrifugal fan is nearly constant at variable speed [39] the fan efficiency at the start of the filtration cycle \(\eta_{fan}(t)\) is the same for both cases of speed controlled fan operation. It can be calculated in Eq. (8) using the actual air flow \(V_U\). The efficiency of the drive was assumed as constant in each case.

3.6. Cost functions

The selected design data for the ventilation systems are summarized in Table 1. The air flow rate is based on the fan inlet conditions. The nominal power of the fan motors was selected by experience. In the calculation of the investment costs only items which are not required in all three configurations were considered. The assembly cost for the different configurations was assumed to be equal for all systems. The typical prices of the optional components are summarized in Fig. 4. The prices for the frequency converters (standard type with B6U rectifier including a built-in DC or AC choke for current smoothing and potential standard EMV) were obtained from ABB and Vacon [40,41]. The prices for soft starters and the additional costs for shielded cables were obtained from Hainzl Industriesysteme [42]. For the dampers the prices were obtained from Aumayr [43]. The price for the flow measurement [44] and the control loop for flow control were estimated at € 1200 independent of the size of the ventilation system. For the cost calculation the linearized cost functions were derived from the available data.

3.7. Chosen parameters for the calculation

For the economic evaluation static methods were selected because of the simple use and the short payback periods expected.
All equations were implemented in an Excel calculation tool, where the payback period for the additional investment was calculated. In the calculation twenty-four-seven operation of the ventilation system was assumed. For a reduced monthly operation time of the system the resulting payback periods have to be adapted by multiplying them by the ratio of 720 to the monthly operation hours. Calculations were carried out for a pressure drop of the air collection system of 1050 Pa and a filter service life of 12,000 operation hours. The ambient pressure was defined as 101.325 Pa. The fan motors were chosen with a motor efficiency class IE2 and a number of pole pairs of two. For the cost of power consumption 0.10 €/kWh was used in the calculations. For the calculation of the increased compressed air consumption of the SCF-CPD operation the following assumptions were used: specific compressed air consumption of filter regeneration: 0.021 Nm³/m²; air to cloth ratio: 1.7 m/min; costs of compressed air: 0.03 €/Nm³.

## 4. Results and discussion

### 4.1. Additional investment cost

The resulting costs of additional investment (CAI in €) in the system with a speed-controlled fan can be calculated by the linearized function $CAI_{SCF} = 46.6 \cdot P_m + 1200$, where $P_m$ is the rated power of the motor in kW. For the system with a control damper the costs of the additional investment result from $CAI_{DC} = 0.87 \cdot P_m + 1330$. As expected, with increasing capacity of the ventilation system the additional cost for the system with the speed-controlled fan increases. In contrast to this, for the system with the control damper the additional investment costs do not depend very much on the size of the system.

### 4.2. Payback period calculation

The results for amortisation of the additional investment costs in the different ventilation system sizes and designs are summarized in Table 2. Generally the payback period decreases with increasing system size. This corresponds with the results for other applications of speed controlled drives [30]. Compared to an automatic damper, for all system sizes the payback period of the speed-controlled fan is shorter. In speed-controlled fan systems the payback period can be cut by approximately half when a controlled maximum pressure drop of the filter cake of 400 Pa is applied. The frequency of filter cleaning calculated as an average for the assumed service life of the filter material of 12,000 h will increase by 80% through such operation. For the calculation of additional compressed air costs, an average cleaning frequency of 0.41/h was assumed for the standard speed controlled fan system. Depending on the system size, the payback period for the additional investment costs of a system with a speed-controlled fan are in the range of 0.7–1.7 years, assuming continuous operation (8000 operating hours per year). This is in a similar range as reported for applications of speed controlled drives in ventilation systems with variable air flow [26]. For two shift operations (4000 operating hours per year), the payback periods are doubled.

The development of the accumulated savings (operating cost savings minus additional investment costs) for a ventilation system with an air flow of 37,500 m³/h and a 55 kW fan motor is shown in Fig. 5.

## 4.3. Comparison of profitability

The profitability, defined as the net profit in a certain period divided by the average tied-up capital, was calculated for the period of the assumed service life of the filter material. A period of 12,000 h is quite unusual as a time basis for the calculation of the profitability. However, it was chosen because the cost savings are not uniformly distributed over the operation time, but are higher for the new filter material and decrease towards the end of the service life. The results are summarized in Table 3. Generally, the profitability increases with increasing system size. The profitability is higher for the speed-controlled system than for the automatic damper controlled system. The highest profitability can be achieved with the speed-controlled system when a controlled maximum pressure drop of the filter cake is applied.

## 5. Conclusions

The use of speed-control systems for fans in small and medium-size constant design air flow ventilation systems which include a regenerative dust filter was investigated. It was shown that the installation of a speed-control system for fans is also an appropriate measure for cost reduction. The use of static methods for the economic evaluation proved to be adequate because of the resulting short payback periods. With the chosen approach, the calculated payback period is between 0.7 and 1.7 years for 24 h operation, depending on the size of the system. Generally, the payback period is shorter for larger systems. For systems in two-shift operations, the payback periods are still in the range of 1.4–3.4 years. The profitability of the installation of a fan speed-control, based on 12,000 service hours, is in the range of 80% to 150%. An operation mode where filter cleaning is triggered by a fixed pressure drop of the filter cake can help to reduce the payback period and improve the profitability further. The system with flow control by an automatic

### Table 2
Payback periods.

<table>
<thead>
<tr>
<th>Air flow rate in Nm³/h</th>
<th>$P_m$ in kW</th>
<th>Payback period in operation hours</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>CD&lt;sup&gt;a&lt;/sup&gt;</td>
<td>SCF&lt;sup&gt;b&lt;/sup&gt;</td>
</tr>
<tr>
<td>9500</td>
<td>15</td>
<td>39,000</td>
</tr>
<tr>
<td>20,000</td>
<td>30</td>
<td>25,000</td>
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<tr>
<td>63,000</td>
<td>90</td>
<td>7600</td>
</tr>
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<sup>a</sup> Constant air flow controlled by a control damper.
<sup>b</sup> Constant air flow controlled by a speed-controlled fan.
<sup>c</sup> SCF with a fixed max. pressure drop of filter cake.

### Table 3
Cost savings in 12,000 service hours and profitability for this period.

<table>
<thead>
<tr>
<th>Air flow rate in Nm³/h</th>
<th>Fan motor size rated power in kW</th>
<th>CD&lt;sup&gt;a&lt;/sup&gt;</th>
<th>SCF&lt;sup&gt;b&lt;/sup&gt;</th>
<th>SCF-CPD&lt;sup&gt;c&lt;/sup&gt;</th>
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</thead>
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<tr>
<td></td>
<td></td>
<td>Cost savings</td>
<td>Profitability</td>
<td>Cost savings</td>
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<tr>
<td>9500</td>
<td>15</td>
<td>400 €</td>
<td>29%</td>
<td>1500 €</td>
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<tr>
<td>20,000</td>
<td>30</td>
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<td>47%</td>
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<td>37,500</td>
<td>55</td>
<td>1400 €</td>
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<td>63,000</td>
<td>90</td>
<td>1800 €</td>
<td>130%</td>
<td>8000 €</td>
</tr>
</tbody>
</table>

<sup>a</sup> Constant air flow controlled by a control damper.
<sup>b</sup> Constant air flow controlled by a speed-controlled fan.
<sup>c</sup> SCF with a fixed max. pressure drop of filter cake.
damper is much less economical compared to the speed-controlled fan system. However, for the assumed conditions it is still better than the uncontrolled system. The installation of speed-control for fans in ventilation systems with a constant design air flow is therefore highly recommended for economic design.

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