584: Advanced Decentralized Ventilation: How Wind Pressure Can Be Used to Improve System Performance and Energy Efficiency

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Abstract

The current paper addresses the problem of existing, highly inefficient ventilation systems with large pressure losses and proposes a new ventilation concept that is in accordance with the lowEx principle. The ventilation system consists of decentralized supply air units and a highly interlaced ducting network. Not only does the conception of this system lead to minimal pressure losses but also allows the wind forces acting on the building to be used for air transportation. Making use of the wind adds additional value to decentralized concepts in that it significantly improves the ventilation performance and reduces the energy costs.
In order to assess the air distribution in the ducting network and the ventilation performance a one dimensional steady state analysis was used. The analysis was carried out for a virtual building with a total ventilation rate of 8000 m^3/h. The steady state analysis showed that in comparison to typical centralized supply system the decentralized system lowers the total fan power demand by a factor larger than 4. Using a 40% overcapacity of fans in the decentralized system leads to a further reduction of fan power demand by almost 25% for the case of wind loading with a wind speed of 10 m/s. The overcapacity of the fans can also be used to decrease cooling load in summer and heating load in winter.

Keywords: exergy, lowEx, decentralized ventilation, hybrid ventilation

1. Introduction

There is a strong need to mitigate anthropogenic emissions of greenhouse gases at present and in the near future. The building sector is a large contributor to the problem with a 38% share of the total primary energy consumption and 34% of the total CO2 emissions world-wide [1]. Buildings have a life span around 50 years such that the cumulated energy consumption for the operation of the building is much more significant than for construction. It is therefore of utmost importance to improve HVAC equipment and reduce the CO2 emission and energy consumption of buildings during operation. An analysis for 13 industrialized countries presented by [2] has shown that space heating has a 61% share on the end use of energy in non-industrial buildings. 53% of this energy is lost through air change. If the average air change was reduced to minimum necessary to insure a good indoor air quality (IAQ) these losses could be reduced to a third of the current level [2]. It is therefore crucial to improve ventilation systems such that good IAQ can be achieved with a minimal energy input. Large saving potentials are available and known as confirmed by [3]. A 70% reduction due to heat recovery, 50% due to effective distribution and 50% due to demand controlled ventilation are possible. Further savings are possible with hybrid ventilation approaches and new control strategies.

2. A Novel Decentralized Ventilation

The ventilation concept presented in this paper is based on a decentralized approach. Instead of having a single supply branch as in a centralized ventilation system, the air is supplied through many fan units arranged along the periphery of the buildings. Decentralized systems have recently gained in popularity especially in Europe. Major benefits are lower pressure drops due to the short transportation distances, space savings and lower construction costs. Balanced, centralized systems are usually not sensitive to varying pressures and temperatures around the building as long as the air intake is placed in an nearly undisturbed region. Conversely, decentralized systems are directly exposed to locally varying pressure and temperature conditions. Increased cooling loads for decentralized ventilation systems due to high temperatures in the vicinity of the façade are usually compensated by lower heating loads in winter due to the same reason [4].
Large projects with decentralized supply systems have been realized in Germany where examples are the Post Tower in Bonn or the Skyline Tower in Munich. In both projects a double skin façade design was chosen to shield the decentralized supply units from direct wind or sun impact. In the Post Tower a highly elaborated, active façade system is used to regulate temperature and pressure distribution in the cavity between the outer and the inner building shell [5, 6]. The supply units are either integrated in the façade or

in a raised floor construction and directly supply the conditioned air to the room on the spot. Unlike these common decentralized supply systems, the system analyzed herein uses a strongly interlaced ducting network to distribute the air over the entire floor space. This network is ideally fed by a large number of supply units and has a sufficiently large capacity. When these two criteria are met, small air flow velocities appear within the network and a nearly homogeneous static pressure distribution can be achieved. Openings then could be arbitrarily distributed across the entire floor space and ensure a sufficient flow of fresh air even to the core of the building. In reality the pressure distribution and the routing in the network are also dependent on the topology chosen. A mesh, in contrast to hierarchical topologies, allows an even distribution of air flows with no need for using large final resistances by throttling the outlets.
The typical control strategy used for decentralized supply is a constant flow rate control. Throttling is used to control the flow rate in case of positive pressure acting on the building. In case of negative wind pressures the flow rate decreases because of the fan's flow rate characteristics unless the fan power is increased. Consequently there are two concurring objectives: Keeping flow rate constant and minimizing power usage of the fans. Allowing the flow rate to vary leads to pressure differences between separated rooms. Conversely, the resisting strategy increases electricity demand for the fans. To resolve this conflict this paper suggests a hybrid approach which tries, besides the mechanical power input, to use wind pressure to enhance air transportation where possible and to cope with non homogeneous flow rates by distribution means. By installing an overcapacity of fans, the redundancy of the supply system is increased and so is the degree of freedom for optimization. A similar ideology can be found in a concept of a hybrid rooftop turbine ventilator presented by [7]. By adding a rooftop ventilator to the classically used DC fan, redundancy of the system is increased and allows for an optimization of the performance with minimal energy input.
Using a hybrid approach for the supply offers the possibility not only to optimize fan power based on pressure boundaries but also to minimize heating and cooling loads based on temperature selective intake of the air. This potential for energy savings will be discussed in this paper based on a 1D steady state analysis of the air distribution. In addition to the first law analysis an estimate of the exergy savings that come along with the energy savings is presented. The exergy losses, i.e. the entropy production due to internal irreversibilities is also addressed. Exergy expresses the high valued part of energy that has the ability to do mechanical work. The minimization of the exergy input for the operation of buildings is hence most important because it minimizes the demand of high grade energy such as electricity or fossil fuels and thereby the emission of greenhouse gases.

3. Method

3.1 Steady State Simulation of airflows

The distribution of the flow rates in the decentralized supply system were analyzed using an adapted one dimensional, steady state formulation of the mass and energy conservation principle for compressible flows. A nodal formulation was used to solve for static pressures in the network nodes. Nodes can be junctions, openings or fans. The mass flows in the pipes where calculated using a simplified equation based on an equation derived by [8] for isothermal flows.

$$p_1^{\,2} - p_2^{\,2} = \frac{16 \lambda R T_s L}{\pi D_h^{\,5}} \dot{m}_n^{\,2} \qquad (1)$$

p1, p2: Nodal pressures
λ: Dimensionless friction factor
R: Specific gas constant for air
T_s: Temperature of the supply air
L: Pipe length
D_h: Hydraulic diameter
m_n: Mass flow evaluated at T_n=293°K and p_n=1e5 Pa

Additional assumptions of an ideal gas and zero elevation were made. For the calculation of the mass flow major and minor losses were considered. Openings also include a loss coefficient for the exhaust into the room. The fans use an additional loss coefficient for the intake through a protection grating in the façade. For the fan characteristics a 2nd order polynomial has been implemented.
As Global input parameters to the simulation ambient pressure of 1e5 Pa and ambient temperature of 273 °K were used. The supply air is assumed to be at a temperature of 293 °K with a specific gas constant of 287 J/(Kg K) and a dynamic viscosity of 1.8e-5 kg/(m s).

3.2 Air Distribution Analysis

A previous study [9] analyzed the air distribution in the floor space for two different topologies: A simple unconnected Fig. 1a and a more complex network topology Fig. 1b. The unconnected topology represents a classical, decentralized supply system. The squares in Fig. 1 represent the fans, the circles the openings and the black diamonds the junctions.

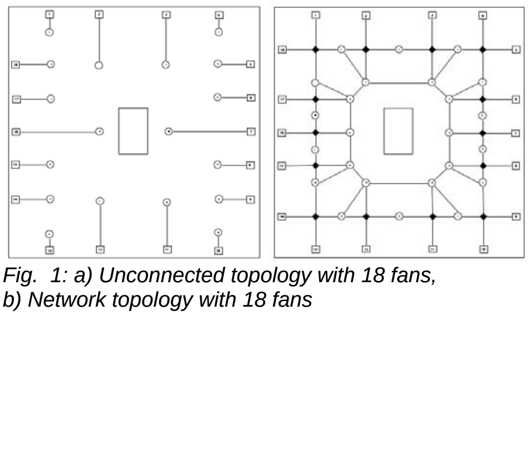

Fig. 1: a) Unconnected topology with 18 fans, b) Network topology with 18 fans

The network topology was generated using different types of network nodes; junctions and openings. Having several openings to be connected in series with the fans was avoided. For that purpose junctions were introduced from where airflows are further distributed to the openings as shown in Fig.2.

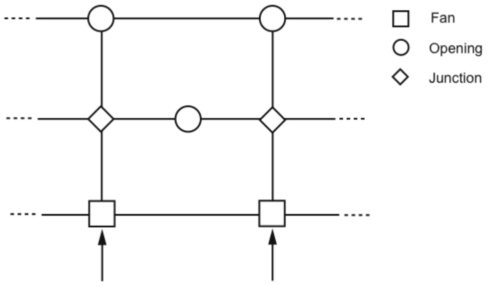

Fig. 2: Connection pattern of fans, openings and junctions

Both setups were studied for a simple square building with five floors and a floor space of 400 m^2 per floor. A specific ventilation rate of 4 m^3/(h m^2) was assumed. 18 fans were placed along the periphery in order to satisfy the design flow rate of 1600 m^3/h per floor. The exhaust air was assumed to be ideally controlled to match the supply flow rate and maintain a constant room pressure. Two cases were differentiated; no wind and wind loading perpendicular to the left sides of Fig. 1a and 1b.

3.2 Analysis of Energy Savings

Based on the above network topology a new network with 26 instead of 18 fans was defined to study potential energy savings possible with this 40% overcapacity of fans. Some of the fans were selectively switched off to reach the same design flow rate as in the previous study.

Similar to the previous analysis a wind loading was defined. The wind speed was selected based on annually wind statistics for Zurich, Switzerland [10]. According to the statistics, values above 6 m/s are probable. A wind speed of 10 m/s was selected. Wind pressures around the building largely depend on the topological circumstances, the orientation and the geometry of the building. Accordingly, the pressure distribution varies strongly along the building envelope and for different heights. For the virtual building defined in this conceptional study it was hence not possible and not of primary interest to define a precise pressure distribution. Instead a simple correlation from [11] was used with averaged pressure coefficients for high rise buildings from Part 4 of the 1995 NBC as described in [12]. Only one situation with wind hitting the building perpendicularly was considered because it is the most critical case in terms of negative surface pressures.

Based on the simulation results for the network under influence of wind, fan power was estimated. The electrical fan power for a fan unit was assumed to be 20 W at full rpm according to [13]. Along a fan curve of constant rpm these fans showed to have an almost constant electrical power demand. The power was then scaled according to the rated rpm using the proportionality laws known for fans [14]. For simplicity a constant total fan efficiency of 20% was used throughout the calculations. The electrical fan power demand of the network was compared to a system with constant flow rate control. Positive pressures were ignored because they are controlled by throttling while for the negative pressures the necessary additional power input was expressed using Equation 2.

$$\Delta \dot{W}_{el,fan} = \frac{\Delta p}{\eta} \dot{Q} \qquad (2)$$

Where η is the total fan efficiency, Δp is the pressure difference and Q dot is the volumetric flow rate. Energy savings due to temperature selective control were also looked at. The reductions in cooling load were calculated according to Equation 3, where n is the number of fans that take in air at a lower temperature $T_{in,2}$ instead of the temperature $T_{in,1}$. The mass flow of one of 18 fans that are simultaneously switched on is expressed by m dot and c_p is the specific heat capacity of air for constant pressure.

$$\Delta Q_{cooling} = n\dot{m}cp(T_{in,1} - T_{in,2}) \qquad (3)$$

3.3 Exergy Analysis

Based on the exergy concept the degradation of high grade energy in the supply air process was analyzed. For this purpose an exergy balance for moist air was applied to the heat exchanger and the fan of the decentralized supply units under consideration. Similar to the analysis of energy savings presented here the influence of variable outside pressure and temperature on the exergy demand was analyzed.

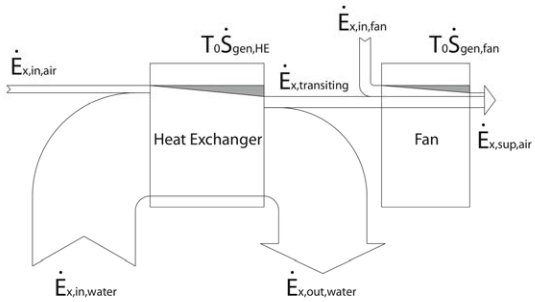

Fig. 3: Grassmann Diagram with exergy flows for the heating case with homogeneous pressure and a local inlet temperature of 12°K above environmental level

The exergy fluxes for the heat exchanger and the fan of a decentralized supply unit are represented in Fig.3 by the Grassman diagram. For the heat exchanger the exergy input originates from the warm/cold water supplied and from the incoming air stream depending on its local properties relative to the ambient condition. The exergy outputs comprise the return water and the heated/cooled air. Exergy destruction appears in the course of heat transfer over a finite

temperature difference. This losses plus the actual exergy load to heat/cool the air constitute the exergy demand of the heat exchanger and must be compensated by the heat generation system. The conditioned air stream is passed to the fan where exergy as electricity is fed and again losses due to entropy generation occur. These fan exergy losses occur due to friction, mixing, etc. and increase with increasing pressure difference applied to the fan. The exergy demand of the fan is basically equal to the electrical power supplied if the exergy flow of the warm/cold air stream is not taken into account and considered as transiting exergy.
As a reference state for the exergy calculation the properties of the external environment were assumed. In the heating case the environment takes a temperature of 273°K, an absolute pressure of 1e5 Pa and a relative humidity of 100%. For the cooling case the same pressure but an environmental temperature of 301°K and a relative humidity of 60% was assumed. For simplicity the state of unrestricted equilibrium with the environment [15] was chosen to be the dead state. Chemical potentials due to the moisture in the air are hence neglected. This simplification only influences the exergy analysis in that it alters the absolute exergy value. The moisture content in the air is accounted for in the calculation of the specific heats and the densities following the rules of psychrometrics. The humid air is treated as a mixture of ideal gases made of air and water vapor. The flow exergy of humid air per kg of dry air [16] writes out as:

$$\dot{e}_{x,mix} = (c_{p,dry,air} + \omega c_{p,vapor}) * \left\{ \left[T - T_0 - T_0 \ln(\frac{T}{T_0}) \right] + T_0 (1+\tilde{\omega}) R_{dry,air} \ln(\frac{p}{p_0}) \right\} \quad (4)$$

where ω is the specific humidity expressed as a mass ratio and ω tilde is the mole fraction ratio between dry air and water vapor. T_0 is the ambient reference temperature and R is the specific gas constant for dry air. The specific heats c_p are expressed for dry air and water vapor.
The flow exergy of the water feeding the heat exchanger is:

$$\dot{e}_{x,water} = c_{p,water} \left[(T - T_0) - T_0 \ln(\frac{T}{T_0}) \right] \quad (5)$$

The exergy balance for any open system is then simply:

$$\sum_i \dot{m}_{i,in} e_{x,i,in} - \dot{E}_{x,consumed} = \sum_j \dot{m}_{j,out} e_{x,j,out} \quad (6)$$

with m dot being the in and outgoing mass flows and e_x dot their respective flow exergies. $E_{x,consumed}$ dot is the consumed or irreversibly lost exergy flow.
The heat exchanger assumed in the energy and exergy calculations has following properties. It has an area of 1.376 m^2 and is layed out for a volumetric flow rate of the water of 0.135 m^3/h using constant supply temperatures of 301°K in the heating and 291°K in the cooling mode. The return temperature was adapted in the calculations to reach a constant supply air temperature of 293°K.

4. Results and Discussion

4.1 Air Distribution And Pressure Losses

In a previous study presented by [9] pressure drops for typical centralized supply systems were estimated and compared to those occurring in the decentralized supply network with 18 fans. As input data the same virtual building as the one considered in this paper has been used. It could be shown that for a total volumetric flow rate of 8000 m^3/h the pressure drop in a centralized supply system is around 800 Pa while for the decentralized supply system it is only 55 Pa, including the pressure drop in the supply unit.
Despite of the higher efficiencies of the large, belt driven radial fans of 70% against 20% of the small, axial DC fans, the huge potential for fan power reductions by a factor of 4.16 was identified using the decentralized supply system.
The analysis of the flow rate distribution for the unconnected and the network topology showed that in case of wind loading a much more uniform distribution can be achieved using a network approach. This holds at least for low wind speeds up to 6 m/s. For a larger wind speed of 10 m/s a significant spread between minimum and maximum flow rates could be observed also for the network. But with 6.06 m^3/h the standard deviation in the network was still significantly lower than the 27.34 m^3/h in the unconnected case [9].
In addition to the open plan situation with the constant room pressure boundary a virtual separation was introduced to represent a single office space in a critical corner surrounded by negative pressure boundaries. In that case a negative pressure boundary of -1 Pa was applied to the openings lying within this virtual office to account for the discontinuity between the biased supply and a constant exhaust. It could be shown that the cumulated flow rate of the openings within the office was only 5.5% below the value for no wind loading. This corresponds to an availability of 94.5% and points out the unique airflow routing qualities of the network distribution system with minute pressure differences [9].
For the analyzed setups shown in Fig.1 the net wind pressure evaluated at the fan locations appeared to be negative. Under these conditions and with the 18 fans available the design flow rate of 1600 m^3/h could not be maintained.

4.2 Energy Demand Reduction

An overcapacity of fans is necessary to be able to maintain the design flow rate. The new network shown in Fig. 4 with 26 fans (40% overcapacity) allows for a flexible allocation of fan power such that it is possible to make positive use of the wind pressure.

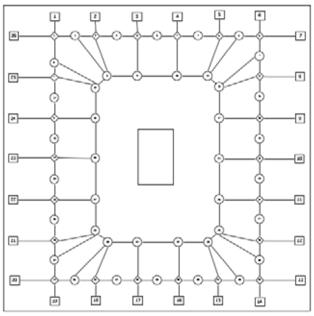

Fig. 4: Network topology with 26 fans

Leaving all 7 fans switched on, on the windward (left side in Fig. 4) and on the opposite side, while switching off 4 fans on both of the other sides where maximum negative pressures appear, allows to reach the design flow rate of 1600 m^3/h. Again only 18 fans remain switched on, such that the total power input remained unchanged compared to the previous study. The reason for this result is mainly due to fact that a different weighting of the wind pressures becomes possible if an overcapacity of fans is available. Also, the number of openings in the network following the topological pattern shown in Fig. 3 over-proportionally increase with the number of fans introduced such that a lower pressure drop per opening occurs.

The electrical power demand of a single fan unit at a prescribed rpm ratio of 82% is 11 W. The 18 fans in use have a cumulated power demand of 198 W.

If a constant flow rate control for the 18 fans is used, an additional power input is needed to compensate for the negative wind pressures. In that case no benefit results from positive pressure because throttling is used to keep flow rate constant. For the wind speed of 10 m/s and the fan allocation according to Fig. 1a the additional power input is 63.8 Watts. In case of constant flow rate control the total electrical fan power is therefore 261.8 W. When a 40% fan overcapacity is added using a switch-off strategy the power demand can be reduced by 25%.

Further energy savings are possible if no effective wind load is present and the overcapacity can be used for a temperature controlled optimization. In summer time large temperatures are recorded on parts of the building that are exposed to direct sunlight. While in summer this creates additional cooling loads it reduces heating loads in winter time. Measurements performed during summertime on three different unbound high rise buildings in Berlin, Germany revealed a maximum difference of 12 °K between ambient temperature and temperatures recorded in the thermal boundary layer, 0.5 cm away from the façade [17].

5.4 °K temperature difference was reported by [18] between opposite sides of an urban canyon in Athen, Greece. Using this temperature differences to estimate the potential reduction of cooling load per fan unit that is switched off on the hot side and switched on a cool side leads to 354 Watts for 12 °K and 159.3 W for 5.4 °K respectively. In the case of an unbound building an overcapacity of 8 fans in the network could be switched off such that a maximum reduction of 2831.5 Watts would result.

4.3 Exergy Analysis

The exergy analysis for the fans under influence of wind leads to the same result as previously calculated using a simple energy analysis. This is because the electrical fan power calculated using Equation 2 is equivalent to exergy, i.e. high grade energy. Using a 40% overcapacity of supply units instead of using constant flow rate control a 25% reduction of the exergy demand for the fans can be achieved under the assumptions made. Much more interesting is the exergy concept for the judgement of the impact of an inhomogeneous temperature distribution around the building on the system performance. For this purpose the exergy balance has to be applied over the heat exchanger. The exergy demand of the heat exchanger directly specifies the amount of high grade energy the generation system has to deliver in order to satisfy the task of heating/cooling. The heat exchanger used in the decentralized supply units is layed out for system temperatures close to room temperatures and is therefore very efficient to use together with a heat pump. If a compression heat pump is used on the generation side the minimization of the exergy demand in the heat exchanging process directly minimizes the electricity input to the heat pump.

During summer time when the ventilation runs in cooling mode the exergy demand for a single unit to cool outside air to room temperature is 7.53 W. When due to sun impact a local inlet temperature of 12°K above environmental temperature is encountered the exergy demand increases by 224% to 24.4 W. In winter when the devices run in heating mode the exergy demand to heat up from environmental temperature is 55.8 W and 42.7% below that value at 32 W for a local inlet condition of 12°K above environmental level. The heating situation with an increased local inlet temperature is exemplified in Fig. 3 with the according scaling. When looking at different configurations and comparing the unconnected topology from Fig. 1a with the network from Fig. 4 with a 40% fan overcapacity and a switch-off strategy, the following exergy savings are observed. For the unconnected topology of Fig. 1a, five devices on the left side of the figure are assumed to be directly exposed to the sun, in winter as well as in summer. For the network topology of Fig. 4, eight devices are completely switched off such that only 18 fan units are running. In the winter case seven devices are switched on on the sunny side, and for summer conditions only devices on the shaded side are running. As a result the winter exergy demand for heating the outside air per floor reduces by 5.4% from 885.5 W to 837.9 W when using the network solution. In summer where the optimization due to switch-off is much more effective exergy savings of 38.4% from 219.9 W to 135.5 W are possible when using the network.

5. Conclusions

As a previous analysis of the flow rate distribution for different topologies showed [9], the network topology leads to an even distribution of the flow rates for small wind speeds up to 6 m/s. Further it was shown that with the slightest negative inside pressure the routing of the airflows can be influenced such that the availability can be maintained also for rooms in critical locations, being surrounded by negative pressures.
In the current study it was found that significant energy and exergy savings result using a decentralized concept with extra savings due to the fan overcapacity installed and the network distribution approach used. The amount of overcapacity determines the potential savings. The larger the overcapacity chosen, the larger the potential savings. Unfortunately, investment costs rise proportionally with overcapacity too. Therefore a trade off must be found between running and investment costs. There are also other factors that limit the amount of overcapacity to be chosen. A minimum of active fans on either side is necessary to satisfy the conditions for an even flow rate distribution. On the other hand there is a limit to the number of fans that can be connected in a certain topological pattern without endlessly increasing the number of openings in the room.

6. Outlook

All the saving potential was examined for a hypothetical building case and it is therefore left to show how the savings would be for a actual building with more rigorous constraints and more realistic wind loadings. On the other hand it would be interesting if the potential savings could be quantified more generally such that these results could be used to estimate the saving potential for an arbitrary type of buildings.
As it was found the energy and exergy saving potential largely depends on the amount of fan overcapacity used. As a consequence a methodology is needed to determine the amount of overcapacity to install. This could be done using statistical weather data. Accounting for probable wind directions and strength as well as for solar irradiation and shading from surrounding buildings in the selection of overcapacity would then lead to an “optimum” performance of the building for most of the time.

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