OPERATING FANS IN PARALLEL IN VIEW OF VARIABLE FLOW RATE OUTPUT
Eddy J. Jacques & Pierre J. Wauters
Department of Mechanical Engineering, Unité TERM, Université Catholique de Louvain, Belgium

ABSTRACT
Starting up a fan or modifying its setting point either by rotational speed variations or blade angle changes, obviously results in new ventilation conditions in the network, which is, of course, the object of the operation. However, since transient phenomena may then occur, particularly in complex networks, there may be a need to simulate these situations prior to any action. This is all the more true, if we are dealing with a multiple interactive fan system.

Simulations with a transient model have shown that the flow in certain airways could be temporarily reversed due to the pressure variations that reach both its ends with a different time lag. This phenomenon may cause dangerous situations to occur in cases of fire or outburst of methane. It is also true that the air flow could be reversed in secondary and less powerful fans, while still rotating in the normal direction.

The study and the experimental investigation of the extension of the reverse flow characteristic is therefore justified from two points of view:
• a numerical simulation model for transient flow conditions must be capable of handling temporary reversed flows, and thus have appropriate fan characteristic curves available over the entire operating domain;
• while operating fans, reversed flows should be avoided or at least reduced. Therefore, the design of start-up procedures or real time control algorithms should take these phenomena into account. A particular interesting application is the operating of two fans in parallel.

In order to investigate experimentally the characteristic curves of a fan under reversed flow conditions a small scale model was built. This facility was designed in such a manner that it was possible to carry out experiments concerning flow reversions and to try out sequential start-up procedures for two fans in parallel.

INTRODUCTION
As a rule, confined or nearly confined volumes where pollutants are produced, or where thermal problems could occur, need to be ventilated. However, in most of these cases, the volume flow demand of fresh air is variable in time. Therefore, the ventilation system must be designed to be flexible, i.e. to be capable of providing the necessary volume flow yet at minimum energy consumption. Fans with variable blade angles or variable rotational speeds are suitable for this job. Whether it would be more economical to work with variable rotational speeds rather than with variable blade angles, or maybe a combination of the two operating modes, is a question on its own and will be of no concern to what is being discussed in this paper.

The easiest solution, if variable ventilation levels are expected, would be to cover the entire range with a single fan. But, in some cases a different solution may be preferred. Setting up two or more fans in parallel may be an attractive solution from the energy saving point of view. If, for instance, during a significant part of the day, the fresh air demand fluctuates only around 50% or less of the peak ventilation capacity, it is obviously advantageous to use a pair of smaller fans connected in parallel. Also from the maintenance point of view, this solution could be justified. But, from the operational and regulating point of view it is not the easiest one. The more that, in a parallel connection scheme, each of these fans has to be fitted with a louver damper, which is, of course, an additional device to be operated.

Mine networks and complex tunnel systems are often ventilated with several fans. One must be aware, though it is not always obvious at first sight, that one or more of these fans, installed at different locations in the network, could show a virtually in parallel operating scheme.

How, in a general manner, a multiple fan system will react and have its impact on the flow distribution in the network is sometimes difficult to forecast, especially during transient situations due to the starting-up or stopping of certain fans. From this point of view, 1D-transient simulations could be a very helpful tool in determining the sequence of operating them.

For a long time engineers knew the basic laws that rule fans in parallel and could draw the resulting characteristic curve (Osborne,1966). But, when it comes to operating them individually in order to obtain a global output volume flow that is gradually and smoothly changing from zero up to some maximum value, literature seems rather poor in giving an answer to the question stated in the next paragraph.

PROBLEM STATEMENT
Suppose a simple case where the volume flow rate $Q$ needs to be adapted continuously between zero and $Q_{max}$. Suppose also that for this application two identical fans are connected in parallel, each capable of a volume flow output:

$$\begin{align*}
Q_1 &= Q_2 = 0.5\ Q_{max}.
\end{align*}$$

The question then is how to operate this multiple fan
system. If one tries to draw, as a function of time, the output flow rate of this fan system that should be the response to a steady or stepwise increase of the volume flow demand, one would not know how to continue the plain curve once the flow rate exceeds significantly the value $Q_1$.

![Figure 1. Variable volume flow rate demand](image1)

With a single fan, the dotted line in the diagram (see figure 1) would represent this further increase of the volume flow rate, but trying to achieve the same operation with a set of two parallel fans makes arise a few questions. They all concern the sequencing of the operations such as opening or closing dampers, starting-up or stopping a fan, changing blade angles or rotational speeds, and so on ...

Formulated in a more general manner, the question is how to operate successively $n$ fans in parallel in order to obtain, at their common discharge point and without creating severe transient perturbations, $m$ different volume flow rates ranging between zero, or $Q_{\text{min}}$ and $Q_{\text{max}}$. $Q_{\text{max}}$ being the sum of the $n$ flow rates $Q_n$, and $m$ being usually much larger than $n$.

Simulations with a numerical model (Jacques, 1995) show that in certain transient situations important reverse flows may occur. Without looking at the quantitative aspects of the obtained results, it is nevertheless obvious that this kind of calculations could not have been achieved if the numerical model did not include a combined characteristic in full of each fan, i.e. a characteristic that extends in the negative flow domain of a fan with its impeller still running in the normal direction.

Unfortunately, very little information exists on the dynamic behavior of a fan if its flow is forced to reverse. Some results with parallel connected fans have recently been made available (Schulze Dieckhoff, 1996). In the present state of the art, it is usually assumed (Douglas et al., 1995), and in analogy with pumps (Hanif Chaudry, 1979), that the reverse flow part of the characteristic may be drawn as the mirror image with respect to the pressure axis, of a passive resistance characteristic to which the static pressure of the fan at zero volume flow rate is added.

Of course, this assumption needed to be verified experimentally.

**FACILITY SET-UP**

Basically, the test facility has a modular design and has been built in such a manner that it could, on minor modifications, serve two purposes. These are, one, forcing the flow in a reverse direction through a fan, and, second, operating two fans in parallel.

![Figure 2. Reverse flow test facility](image2)

Figure 2 shows the layout of the test facility in a first set-up i.e., the reverse flow set-up. The essential elements are:

- Three airways, each connected to a chamber C. They form a network of three branches, B1, B2 and B3;
- Fan F1 which is the axial fan under study in reverse flow conditions. It is located in the first branch, B1, and its power is approximately 2 kW at 3000 rpm;
- Regulator R in branch B2. Its main function is to control the volume flow in B1, but it acts also as a sort of security valve in case fan F1 starts to stall;
- Fan F3, a more powerful fan of around 5 kW at 3000 rpm, is mounted in a larger duct. It has to be considered as an external fan, its only goal being to force a flow throughout the entire circuit in a reverse direction.

The different fan characteristics, and consequently also the flow rate variations, are obtained by changing the frequency of the motor of fan F1, and by using a Ward-Leonard group for moving fan F3. The entire test facility has been dimensioned in order to comply with the ISO requirements for measuring flows and pressures, at least when the air flows in what is considered to be the normal direction.

A slightly different layout of the same test facility is achieved in a second set-up, which is the parallel fan connection. Figure 3 shows the essential parts:

- Three airways forming the same basic network with three branches B1, B2 and B3. The latter one is a large, slightly divergent duct ending with a flow straightener and a short duct with an outlet orifice plate;
- Two similar, though not identical fans F1 and F2, are connected in parallel by means of two ducts of the same internal diameter. Each fan is equipped with a flow straightener, and further upstream, with in-duct orifice plates;
- Regulator R at the inlet side of fan F2.

![Figure 3. Test facility with fans in parallel](image3)

**MEASUREMENT INSTRUMENTATION**

The design of the test facility has primarily been guided by the concern to comply, when possible, with the standard procedures for measuring pressures and flow rates. 'When possible' means: if the air flows in what is called the normal direction with regard to the fans. The volume flow rates are then measured by orifice plates.

However, since the specific aim of the test facility is to determine the fan characteristic in reverse flow conditions, it is obvious that in parts of the circuit and during certain tests, the standards techniques would not be applicable. Therefore, Pitot tubes have been used with all the inconveniences thereof such as traversing problems, velocity profiles investigations, finding the right measurement locations, and so on...

Static pressure is measured by a technique of averaging over four tappings, diagonally located.

Direct measurements of the rotational speeds of the fans have been carried out with a revolution counter technique.
EXPERIMENTS AND RESULTS

Preliminary tests

Since a great deal of the tests had to be conducted in non-standard measurement conditions, it was believed that checking the balance of the three independently measured flow rates at their junction, would give a good indication about the quality of the chosen measurement technique. This verification has been carried out for three different flow configurations. The results showed a closing error of the volume flows that did not exceed 4% of the merging flow, and therefore, allowed us to consider that the tests could be carried out in a satisfactory manner and that the measurements would be valid.

Reverse flow

Submitting a fan to a forced reverse flow could lead to unfortunate consequences. In order to safeguard this as much as possible the fans, the tests have been conducted with a lot of care. The following figure 4 may show the sequence applied for gradually increasing the flow rate in the circuit, so realizing the reverse flow through fan F1.

Figure 4. Sequence of the flow rate increase in the circuit

Keeping the rotational speed of fan F1 as close as possible at 300 rpm, a first set of tests with different flow rates enabled one to plot the corresponding negative part of the characteristic of fan F1 (see figure 5). The same kind of tests were then repeated for two more rotational speeds which are 600 and 900 rpm respectively. These characteristics have also been drawn in figure 5.

Figure 5. Fan characteristics in full at 300, 600 & 900 rpm

At this point a few remarks need to be made:
- the experimental results around the origin of the axis are missing. The reason is that the volume flows close to zero could not be measured validly;
- the different duty points are obtained basically by changing the rotational speed of fan F3 and, in a few cases, by operating concurrently the regulator R. This operation causes the equivalent resistance of the entire circuit to be changed, which results in the rotational speed of fan F1 moving slightly away from its set value, in this case 300, 600 and 900 rpm. In order to obtain an homogeneous set of pressures and volume flow rates with respect to the same rotational speed of e.g. 300 rpm, the measured values have been corrected according to the similarity laws.

The curve of 300 rpm shows clearly the quadratic law that rules the pressure as a function of the volume flow rate. This observation is maybe less obvious for the higher rotational speeds of fan F1 for which, due to power limitations of fan F3, the available volume flow measurement domain was less extended.

Another interesting experimental result shows that the relationship between the rotational speed and the volume flow rate which is known to be linear, keeps more or less this linear behavior in the negative flow domain, notwithstanding the fact that the flow must be very turbulent.

The closing error $e$ on the flow rates at the three branch junction has been calculated for each test. The evaluation formula:

$$ e = \frac{(Q_2 + Q_3 - Q_1)}{Q_1} $$

applied on the eighteen test cases, yields an average value of around 1.9%, a standard deviation of 1.6% and a maximum value of 6.4%.

Fans in parallel

One of the problems in operating parallel fans is to determine the right time when the second fan needs to be started up because of an additional volume flow demand. As previously explained, this operation should be carried out in such a manner that the total output flow rate increases gradually, yet as fast as possible. Therefore the tests begin with fan F1 already running at its design speed, which is roughly 2940 rpm at a motor frequency equal to 50 Hz. Three kind of tests are considered:

Set 1

Since actions have to be taken concurrently on the
rotational speed of the fans and on the opening of the regulator, a first set of tests have been carried out to observe how the volume flow rates would evolve in branches B1, B2 and B3 if only the regulator is operated. These tests have then been repeated for several rotational speeds of fan F2. Figure 6 shows these parametric curves of the volume flow rate in branch B2 as a function of time. The total time that is necessary for the regulator to open, from a closed position to a full aperture, is approximately 90 s.

Further to this, one can see that it is impossible to avoid completely any flow reverse. However, this reverse volume flow moved through fan F2 is small, less than 4 % of its design capacity. The simulation also allows one to observe that the volume flow discharge of fan F2 becomes zero after 4 s. This point, defined as Z, corresponds in this case with an opening of the regulator that is roughly 5 % and a rotational speed of nearly 1250 rpm. Within these 4 s, the total volume flow discharge seems not to suffer from any perturbation. The attention should nevertheless be drawn to the fact that the time span Z is dependent on the equivalent resistance of the circuit, and therefore has to be evaluated prior to the design of the actual operating procedure.

Set 3.
Operate both fans simultaneously in order to achieve the required flow rate $Q_3$ which is, of course, assumed to exceed significantly the maximum flow rate of a single fan. It is obvious that the resulting flow rate $Q_3$ can be obtained by several combinations of the flow rates of fans F1 and F2. Assuming that the exceeding demand is for example 50 % more, one of these combinations could be:

$$Q_3 = Q_{1,max} + 0.5 Q_2.$$  

However, it can be theoretically demonstrated that sharing the load equally over both the fans F1 and F2 is, from the energy point of view, preferable to any other workable combination. The results of tests with fan F2 moving at eight different rotational speeds and F1 at its design speed, are plotted in figure 8. They confirm experimentally the statement that it is always more efficient, and thus more economical, to have both fans running at the same rotational speed. Therefore, the control procedure should also include the setting of the motor frequencies of both fans at identical values. This operation means in this case that the rotational speeds of both fans should be changed in opposite directions, i.e. increasing one and decreasing the other, in order to reach as fast as possible the ultimate duty point where both fans run at the same rotational speed.

**OPERATIONAL PROCEDURE**
The results of the different aforementioned tests enables one to formulate an operational procedure that could be included in a local controller of a twin fan system. Once the point Z has been determined, the procedure would look as follows:

1. when the second fan needs to be put in operation, synchronize the opening rate of the louver damper with the increase of the rotational speed of the fan motor in order to reach as fast as possible the defined point Z. Then speed up the fan motor while going on with the opening of the louver damper till the required volume flow rate at the common discharge point is reached;
2. operate both fans simultaneously by changing their motor frequencies or their blade angle settings in an opposite direction in such a manner that the total flow rate remains constant. This step needs the time diagrams to be known, i.e. the variation rate of either the rotational speed or the blade angle as a function of time, because these curves might not be linear.

Of course, for maintenance reasons, the logic position of fans F1 and F2 can be switched, but if they are not really identical, the point Z may correspond to slightly different values for the angle setting of the louver damper and the rotational speed of the second fan.

It must be emphasized that no thought has been given in this paper to the special case of combining fans with characteristics having points of contraflexure with the result that several flow rates are possible at a single value of the fan pressure.

CONCLUSION

The experimental work and the results described in this paper are twofold:
• they contribute to give experimental evidence of what had been assumed to be the fan characteristic in full, based so far only on analogies with pumps;
• they enable one to establish a few guide-lines for operating fans in parallel.

Although the work has been carried out on a small scale model, there seems to be no reason a priori why these results could not be extended to larger fans. Nevertheless, the work could be completed by:
− more detailed investigations about the energy saving aspects with much larger fans;
− connecting parallel fans of which one is much smaller than the other. This aspect would certainly put the problem of energy saving in a different manner.

REFERENCES


