Experimental investigation of an actively controlled automotive cooling fan using steady air injection in the leakage gap

Tarik Azzam[1,2], Richard Paridaens[1], Florent Ravelet[1], Sofiane Khelladi[1], Hamid Oualli[2], Farid Bakir[1]

Abstract

In an axial fan, a leakage flow driven by a pressure gradient between the pressure side and the suction side occurs in the gap between the shroud and the casing. This leakage flow is in opposite direction to the main flow and is responsible for important energy dissipations. Therefore, many authors have worked to understand this phenomenon in order to reduce these inherent energy losses. Up till now, most of the studies reported in literature are passive solutions. In this paper, an experimental controlling strategy is suggested to reduce the leakage flow rate. To this aim, a fan with hollow blades and a specific drive system were designed and built for air injection. Air is injected in the leakage gap at the fan periphery. The experiment was performed for 3 rotation speeds, 5 injection rates and 2 configurations: 16 and 32 injection holes at the fan periphery. The experimental results of this investigation are presented in this article.

Nomenclature

D diameter radius (m)

Gain power gain added by the control

P power delived by the fan (W)

P_0 power delived by the fan for $q_{inj} = 0$ (W)

q_{inj} injection rate (m^3.s^{-1})

q_{max} maximum fan flow rate (m^3.s^{-1})

q_v fan flow rate (m^3.s^{-1})

R_e fan Reynolds number

R_{int} hub radius (m)

R_{max} tip radius (m)

x_{fan} relative position between the fan and the carter (m)

x_{holes} position of the injection holes on the casing (m)

W relative velocity of the fan (m/s)

ΔP pressure difference generated by the fan (Pa)

μ air dynamic viscosity (Pa.s)

φ flow coefficient

ψ static pressure coefficient

ρ mass density (kg.m^{-3})

Introduction

The leakage flow occurring at the blade tip has a strong impact on turbomachinery. This flow driven by the pressure difference between the pressure side and the suction side is responsible for important energy dissipations and sound generation. This is certainly the reason why a huge number of authors are involved in such a problem investigation and sought mainly to enlight the underlying mechanisms behind this phenomenology. Boudet et al. (1) investigated experimentally and numerically the tip leakage flow in an axial fan. They focused on the tip leakage vortex wandering. This vortex appears to be of major contribution in noise radiation. Fischer et al. (2) measured the flow in

[1]DynFluid, Arts et Métiers, 151 boulevard de l'Hôpital, 75013 Paris, France
[2]Ecole Militaire Polytechnique, Laboratoire de Mécanique des Fluides, Algiers, Algeria

Corresponding author:
Richard Paridaens, Arts et Métiers, 151 boulevard de l'Hôpital, 75013 Paris, France
Email: richard.paridaens@ensam.eu

the tip gap of a turbine using an optimized non-intrusive measurement concept based on frequency modulated Doppler global velocimetry. They succeeded to resolve the vortex structure of the tip leakage flow the By with high temporal resolution. Pogorelov et al. (3) numerically investigated the vortical structures around the tip-gap region of an axial fan. The authors showed that the reduction of the tip-gap size totally changes the flow field. It decreases the amplitude of the tip-gap wandering vortex resulting in a significant sound power reduction. The effects of tip-gap size on the tip-leakage vortical structures are investigated by You et al. (4) using large-eddy simulation. Their results suggest that the tip gap size has a strong impact on the tip-leakage cavitation leading to noise and vibration enhancement and thus degenerative performances. Sang Woo Lee and Seong Eun Lee (5) investigated the impact of the squealer rims in a turbine cascade. By modifying the height-to-span ratio, the total-pressure loss coefficient mass-averaged could be reduced by a rate of 11.6%.

To reduce the leakage flows, different geometrical solutions have been investigated. Many of them deal with the optimization of the labyrinth seal (6; 7; 8; 9; 10). Wallis (11) tried to control the leakage flow in a turbine by using turning devices. Corsini et al. (12; 13) investigated the use of profiled endplates at the blade tip and succeeded to change the orientation of the leakage flow structure. AktÃijrk et al. (14) studied the impact of five different tip platform extensions on the blade pressure side. Pardowitz et al. (15) used a rotating shroud band to suppress the tip clearance vortex system. Longhouse (16) showed that rotating shroud band is a practical solution for controlling tip clearance noise. In electric motor, the cooling airflow is generated by a radial flow rotor. Vad et al. (17) redesigned the existing fan rotor to a shrouded axial one. For the same cooling performance, they reduced the fan noise and the power consumption. Gourdain et al. (18) considered rotor-stator interactions and tip leakage flow effects on overall performance and aerodynamic stability. The authors investigated 3 configurations. The configuration 2 has a larger clearance gap compared to the configuration 1. The configuration 3 is identical to the configuration 1 with a special casing treatment based on a honeycomb design. Comparing the configuration 2 to 1, the authors showed that with a larger clearance gap the stable operating range could be reduced by 40% and the efficiency by 2%. When comparing the configuration 3 to the configurations 1 and 2, the authors succeeded to obviate the losses with the special casing treatment.

In automotive, the leakage phenomenon strongly impacts the global performances of the cooling fan (19; 20; 21). Thus, due to confinement constraint, the leakage gap is larger than in other area: the gap to blade height is about 2% (22). This paper deals specifically with the phenomenon of leakage flow in an axial automotive cooling fan.

Because of numerical geometry refinement limits, this article investigates the control by tip injection. Many authors already used this solution to improve turbomachinery characteristics. Using the injection upstream of the rotor, Weigl et al. (23) succeeded in stabilizing rotating stall and surge in a transonic compressor which provided an enlargement of the operating range. Rhee et al. (24) increased the lift of a hydrofoil by injection control. Eberlinc et al. (25; 26; 27) experimentally increased the pressure of an axial fan hollow blades by a rate of 6%. This gain was due to a free jet applied at the trailing edge of the blades. This free jet flow led to flow velocity increasing and adverse pressure gradient decreasing in the boundary layer. Wang and Zhao (28) experimentally and numerically investigated a transonic turbine cascade with different trailing edge ejections. As Eberlinc et al. (25; 26; 27), the authors successfully reduced the vortex at the trailing edge of the blades.

In this paper the control by tip injection is used to reduce the phenomenon of leakage flow occurring at the blade tip. Neuhaus and Neise (29) attempted to reduce the leakage flow of an axial fan applying an active flow control at the tip region of the impeller. Air is injected in the opposite direction of the flow by slit nozzles flush mounted in the inner casing wall. With this process, the authors were able to shift the stall point towards lower flow rates and to enlarge the useful range of the fan characteristics by 62%. They could also increase the pressure by 28% and the efficiency by about 10%. Morris and Foss (30) study was devoted to relatively large (2.5 cm) tip clearance required in automotive cooling fans. A jet of high momentum air was delivered by a fixed shroud into the leakage gap. Enhancement of both the performance and efficiency could be ensured at higher flow rates. Using a similar aerodynamic fixed shroud concept, Neal and Foss (31) investigated the impact of air injection into tip clearances smaller than those of Morris and Foss. The authors increased the flowrate by 34% and the efficiency by 17% with the air injection system. Niu and Zhang (32) measured the effect of an active tip-clearance control method based on tip injection in a high-turning axial turbine cascade. The authors injected air through a row of 10 equidistant holes, directed towards the pressure surface at an angle of $45°$ relatively to the blade tip surface. They found that tip injection can effectively reduce tip clearance mass flow, resulting in a smaller tip clearance vortex and lower losses in the passage. In order to reduce the blade tip clearance leakage in turbine designs, Wadia et al. (33) numerically investigated the effects of an active jet-flow injected into the leakage gap for five differently located injection holes and three jet-flow mass rates. The efficiency could be enhanced by an amount of 0.35% with loose tip clearance, and by 0.3% with tight tip clearanceHamik and Willinger (34) analytically and numerically investigated the concept patented by Auxier (35) in 1995. The idea consists into connecting the blade leading edge to the

blade tip by an internal channel. Due to the pressure difference, a jet is injected to the tip gap and blocks the leakage flow. With an injection mass flow rate of 0.5% of the mass flow, they predicted a performance improvement which was independent of the tip gap width. Hu et al. (36) applied the concept proposed by Auxier to an axial fan and experimentally investigated the interaction between the tip leakage flow and the tip injection flow using a 2D-PIV technique. As expected, the authors observed a reduction of the leakage flow. However, the tip injection generated a wake flow phenomenon similar to the Karman vortex street resulting in more intricate flow with non-stationarity enhancement.

In this study, an active control by air injection into the clearance gap between the shroud and the casing of an axial automotive cooling fan is experimentally investigated. The air is injected in the gap between the rotating shroud and the fixed casing. It is sought to to reduce the leakage flow. The fan with hollow blades is built using a rotomolding process. The air is injected through the fan axis and excited at the shroud ring in order to counter the leakage flow. Interesting results are obtained by this air injection controlling technique.

Experimental setup

The axial-flow fan with hollow blades

The fan (Figure 1) used in this study is an automotive cooling fan. It is a controlled vortex designed axial-flow fan with thick and hollow blades (37). The hub-to-tip radii ratio (R_{int}/R_{max}) of the six blades is equal to 0.337. The tip radius (R_{max}) is equal to 179 mm. This fan is in polyethylene material and obtained using rotomolding process. A suited mold is loaded with polyethylene powder (500g) and heated to 250°C. System rotation disperses the softened material sticking to the walls. A hollow and circular shroud ring of 31-mm axial length and 9-mm thick is added to the fan. Adjunction of this shroud ring at the blade tips significantly impacts on the loss mechanisms as reported in (33; 38; 39). For brevity, the effect of such a shroud used to ensure stronger mechanical properties is alleviated but details can be found in references (33; 38; 39). Naturally, the hollow character of the fan is dictated by the need of inside fluid circulation.

With this internal flow circulation, different applications can be met. It is already intended to investigate the impact of injected air at the trailing edges (Figure 2a) as extension of the present work. In fact, it is reported in literature that injecting air at this position locally increases the flow velocity with tendency to reduce the boundary layer
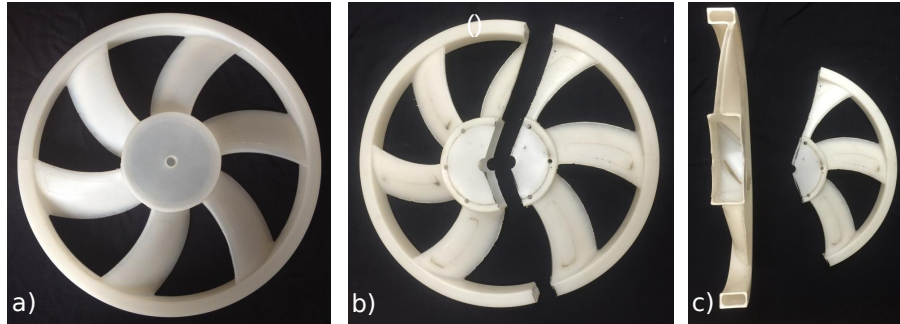
Figure 1. The hollow fan: the whole fan (a) and the fan cut in two parts (b,c).

stalling, Eberlinc et al. (25). It is also planned to experiment injection at the side of the shroud as shown on figure 2b. This is expectedly to reduce leakage flow. In this study, air is injected into the fan and exits at the periphery of the shroud (Figure 2c) in a 4-mm leakage gap. Two configurations are explored: the shroud is composed of 16 or 32 injection holes of 4 mm regularly spaced on the circumference. The holes are oriented in the direction $(r, \theta, z) = (1, -1, 1)$ in such a way to counter the leakage flow. The direction has a component in θ to compensate the fan rotation.
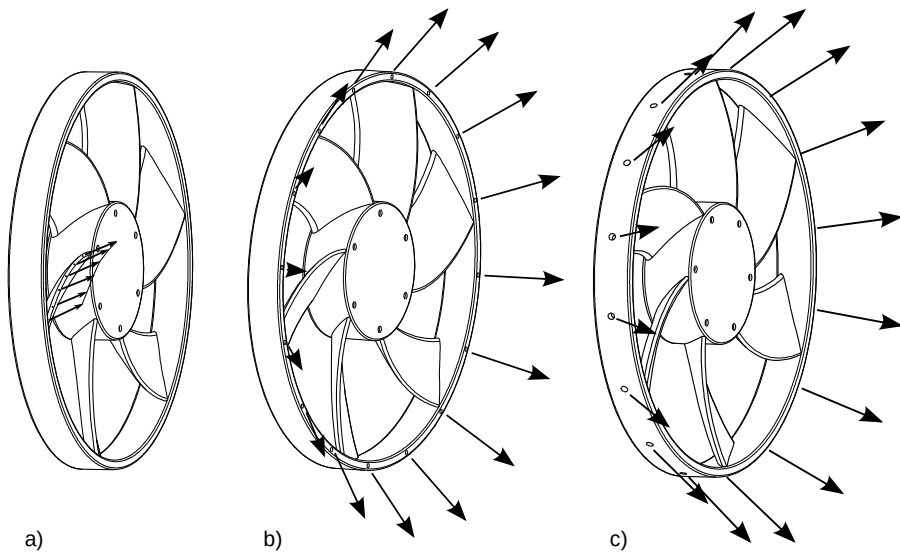

Figure 2. Actual and further investigations: injection at the trailling edge of the blade (a), injection at the side of the shroud (b) and injection at the periphery of the shroud (c).

The carter used in this study is of the same thickness as the fan: 46 mm. Thus, many positions between the fan and the carter are possible. The figure 3 (a) represents 3 relative positions: the output position with an offset of -30 mm between the fan and the carter, the standard position with no offset and the retracted position with an offset of +30 mm. This relative position has a strong influence on the fan characteristics especially when the injection control is operating. In a similar way, the position of the injection holes on the casing also has a strong influence on the control. The figure 3 (b) represents 3 different positions of the holes on the casing $x_{holes} = 1/4L$, $x_{holes} = 1/2L$ and $x_{holes} = 3/4L$. The impacts of those positions are not investigated in this article. The results presented here are obtained for a standard position between the fan and the carter ($x_{fan} = 0$) and for the injection holes located at the middle of the casing ($x_{holes} = 1/2L$).
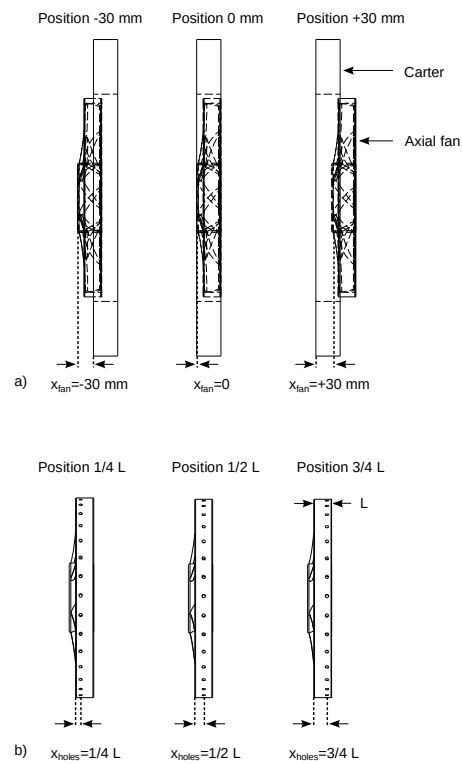

Figure 3. The relative position between the fan and the carter (a) and the position of the injection holes on the casing (b).

The drive system

To supply air to the fan and the casing, the 3-bar compressed air system of the laboratory is used. A flowmeter and a valve are installed upstream the fan and the casing in order to control the injection. For fan security reasons, the injection is made at the maximum injection flow rate equal to 700 L/min at an unknown low pressure. Yet, it is planned to strengthen the fan by using another material rather than polyethylene. A specific drive system shown on Figure 4 was built to connect the compressed air system to the hollow fan. The drive system has two functions. It transmits the rotation from the motor to the fan and allows the injection from the compressed air system into the fan. To fulfill its functions, the drive system is composed of two shafts connected through a pulley and belt system. The first shaft connects a 4-kW motor to the pulley and belt system. On this shaft a HBM strain gauge transducer is installed. It measures the torque with an uncertainty corresponding to 0.1% of the full scale. In this case the full scale is equal to 5 N.m. The second shaft connects the pulley and belt system to the fan. It also connects the compressed air system to the fan. This connection is possible because the shaft has the particularity to be hollow. Into the shaft, a rotary joint is used to inject air from a stationary source to a rotating piece. The rotary joint is brand Deublin and has the reference 1115-000-200. The fan angular velocity is measured with a tachymeter of relative precision $\pm 0.2\%$
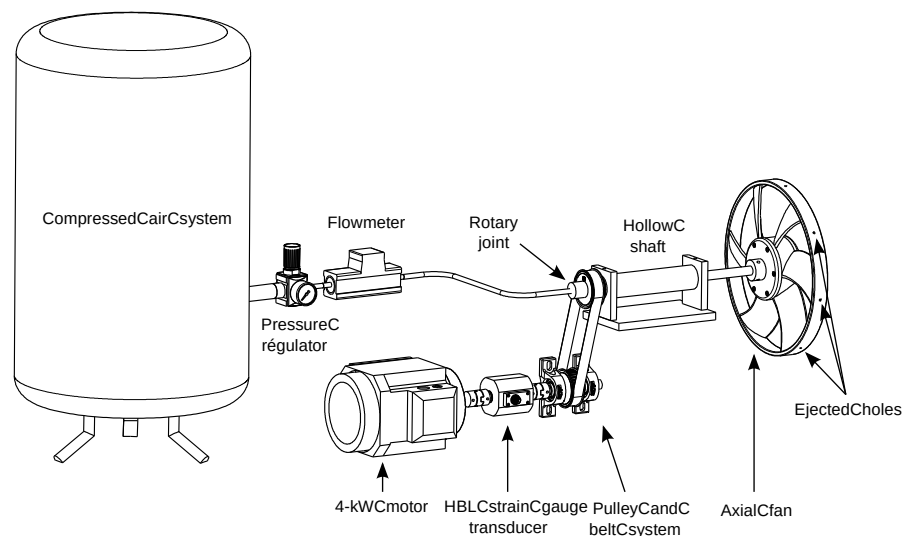

Figure 4. The fan drive system.

In this article, air injection by a compressed air system is used as convenience. The investigation does not focus on how the air injection is made. It rather focuses on the potential contribution of the control by air injection in an axial fan. Yet, in a further study the authors project to fix an internal fan to the main one in order to generate the air injection in a passive way. This system described on Figure 5 is composed of the main axial fan and an internal centrifugal fan. The internal centrifugal fan is fixed to the axial fan and has the same rotational speed.
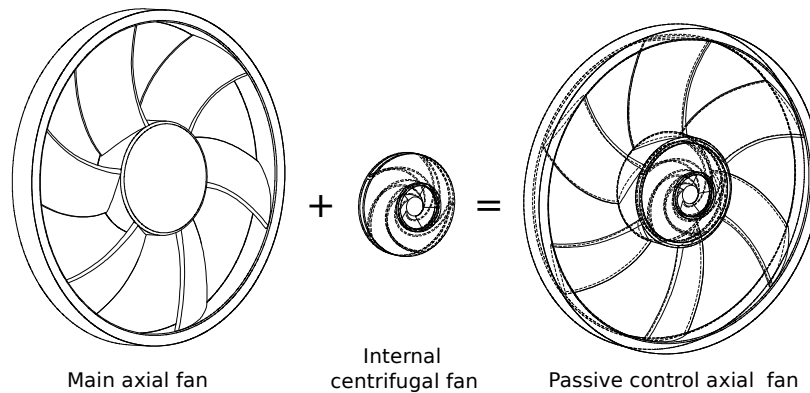

Main axial fan   Internal centrifugal fan   Passive control axial fan

Figure 5. A passive control axial fan.

The test bench

To determine the global performances of the fan, an air suction test bench was used. This experimental facility shown on Figure 6 was designed and built at the laboratory according to the ISO 5801 standard [10]. It consists of a cuboid of size 1.3 x 1.3 x 1.8 m. A pierced plate inside it prevents flow prerotation. The air flow rate is set and measured according to the ISO-5167 norm [11] by setting the aeraulic resistance of the bench through diaphragms of various sizes. The elevation pressures are measured with an absolute precision of ± 0.1Pa.

Results

The experiment was carried out for 3 rotational speeds, 5 injection rates and 2 configurations with a different number of injection holes at the fan periphery. The results are presented on Figure 7. The figure represents the fan characteristics for a rotational speed equal to 1000 rpm (a, d), 1500 rpm (b, e) and 2000 rpm (c, f), and for a
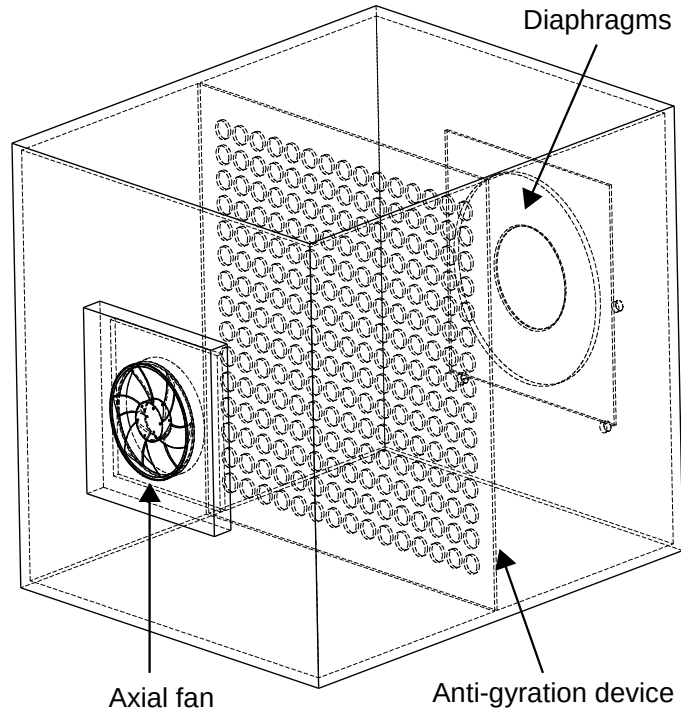
Figure 6. ISO-5801 test bench. The dimensions are 1.3 x 1.3 x 1.8 m. The fans suck the flow through the test bench.

number of injection holes equal to 16 (a, b, c) and 32 (d, e, f).ψ and φ are respectively the static pressure and the flow coefficient:

$$\psi = \frac{2\Delta p}{\rho u^2} \tag{1}$$

$$\varphi = \frac{4 q_v}{\pi D^2 u} \tag{2}$$

In order to quantify the impact of the control by air injection, the fan characteristics are plotted for 5 different injection rates. These are normalized and reported to the maximum fan flow rate. The maximum fan flow rate depends on the rotation speed and it is equal to the rate when the static pressure difference is zero for the configuration with no injection. The maximum fan flow rate is respectively equal to $0.39 m^3/s$, 0.67 m^3/s and $0.97\ m^3/s$ for corresponding rotation speeds equal to 1000 rpm, 1500 rpm and 2000 rmp. The black curves represent the fan characteristics with no injection. These are identical for 1500 rpm and 2000 rpm and different for 1000 rpm. This

difference probably comes from additional losses appearring at low Reynolds number (40). The Reynolds numbers are respectively about $1.20 \, 10^5$, $1.80 \, 10^5$ and $2.40 \, 10^5$ for a rotation speed equal to 1000, 1500 and 2000 rmp at the design flow rate, $\varphi = 0.17$ The Reynolds numbers are defined as follow :

$$R_e = \frac{\rho W R_{max}}{\mu} \tag{3}$$

At low flow rate, the injection does not modify the fan characteristics regardless of the rotational speed and the number of holes. When increasing the flow rate, however, the pressure coefficient increases as well. This is well observable when flow rates become sufficiently high (29). As expected, the pressure coefficient increases with the injection rate. Nevertheless, a saturation phenomenon can be observed. For example, on Figure 7 (a), an injection rate equal to 6% does not seem to affect the pressure coefficient comparatively to an injection rate equal to 5.1%. It can be conjectured that this phenomenon appears when the leakage flow is drastically reduced; so that an additional injection rate is useless. At high rotation speed, the control by injection seems to be less significant. However, this result is to be linked with the value of the normalized injection rate which decreases with the rotation speed. At 1000 rpm, the injection rate is equal to 6% of the maximum fan flow rate while at 2000 rpm the ratio is only 4.6%.

To analyze the influence of the number of holes, the figure 8 is plotted. On this figure, the power gain added by the control versus the normalized injection rate are presented for rotation speeds, 1000 rpm (a, d, g, j, m), 1500 rpm (b, e, h, k, n) and 2000 rpm (c, f, i, l, o) and for the 5 throttles. This gain is defined as follow:

$$\text{Gain} = \frac{P - P_0}{P_0} \tag{4}$$

$P = q_v \Delta P$ is the power delivered by the fan and P_0 is the power delivered by the fan for $q_{inj} = 0$ Each throttle corresponds to a different diaphragm of the test bench (Figure 6) and so it corresponds to a different aeraulic resistance of the circuit. The resistance increases with the number of throttle. For each figure, the results obtained for the 2 configurations are represented. The results for 16 and 32 injection holes are respectively plotted with circle and cross markers. In this study, the active control by air injection adds an increase up to nearly 40% in the delivered power. For an identical normalized injection rate (q_{inj}/q_{max}) the most important gains were obtained for the lowest rotation speed (1000 rpm) and the lowest aeraulic resistance (throttle 1). As expected, the delivered power does not change between the 2 configurations when there
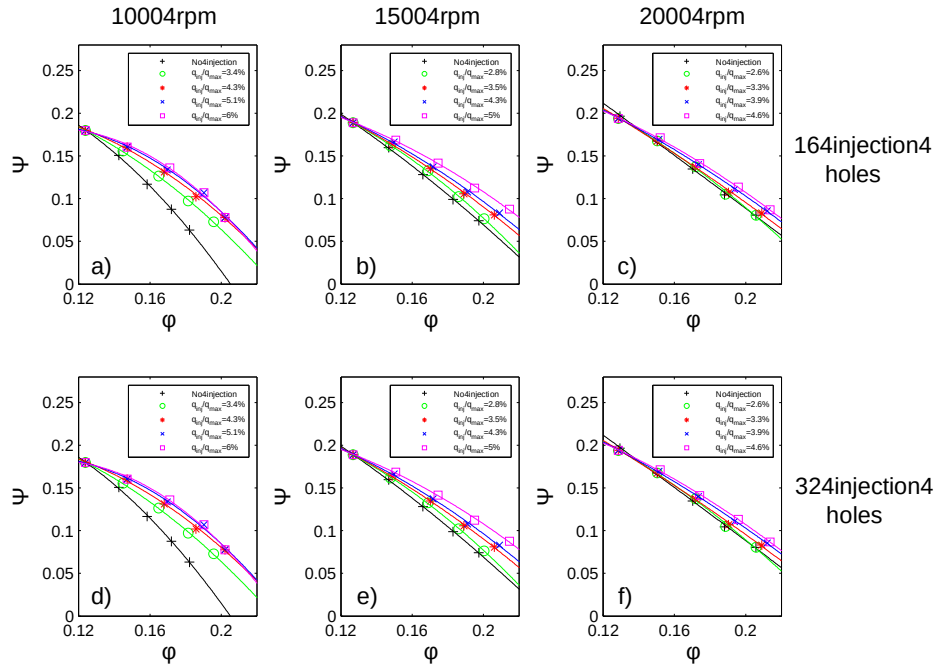
Figure 7. Comparison of aerodynamic characteristic of the axial fan for 5 different injection rates and for 3 rotation speeds: 1000 (a, d) 1500 (b, e) and 2000 (c, f) rpm and for 16 (a, b, c) and 32 holes (d, e, f) at the fan periphery.

is no injection. However, for the same injection rate, the delivered power is higher for the configuration with 16 holes than for the configuration with 32 holes. This result may be understood by referring to the velocity injection at the periphery of the fan. By mass conservation, for the same injection rate, the configuration with 16 holes has a velocity injection twice higher than the configuration with 32 holes. Thus in this case, a higher velocity injection spread across 16 holes seems to better reduce the leakage flow than a lower velocity injection spread across 32 holes. Regardless the number of holes, an increase of the injection rate increases the power gain. However, the saturation phenomenon could also be observed in this Figure especially at high power gain (a, d) and at low working flow rate (k, l, m, n, o). When this phenomenon appears, an increase of the injection rate and the modification of the number of holes have no influence on the control. The authors think that this phenomenon appears when the leakage flow becomes nonsignificant.
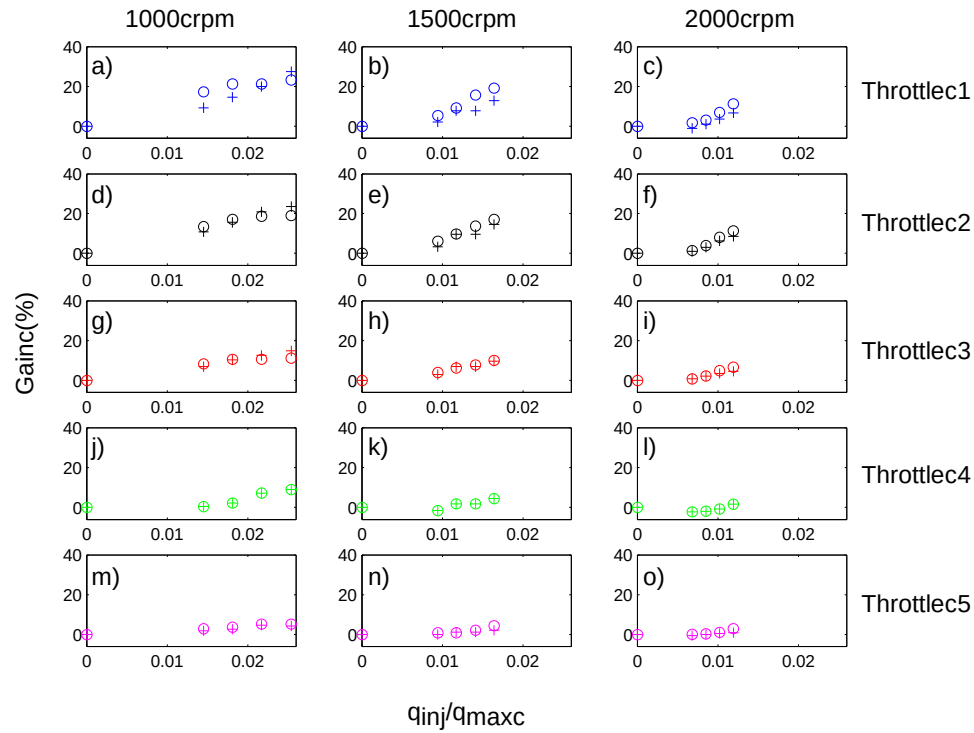
Figure 8. The power gain added by the control for 3 rotation speeds: 1000 (a, d, g, j, m), 1500 (b, e, h, k, n) and 2000 rpm (c, f, i, l, o) and 5 throttles. +: results for 16 injection holes. o: results for 32 injection holes.

Conclusion

A rotomolding process was used to build a hollow fan in order to study the leakage flow control by air injection at the fan periphery. To drive the fan and simultaneously control the air injection, a specific drive system was designed and built. Experiments were carried out and conducted for 5 injection rates, 3 rotation speeds and 2 configurations, 16 and 32 holes at the fan periphery. This control leads to an improvement of the fan characteristics. An increase up to nearly 40 % in the delivered power was registered. In fact, by reducing the leakage flow, the blades could work better especially at the periphery. The experimental results show the control is particularly efficient at high flow rate. An increase of the injection rate generates higher pressure coefficients. In this case, for the same injection rate, better results are obtained for the configuration with 16 holes at the fan periphery than the configuration with 32 holes. A saturation phenomenon is also observed and is likely due to weak leakage flow. As perspectives, the authors plan to consider the impact of air injection at the trailing edges. Injecting air

at the trailing edges increases locally the velocity flow and tends to reduce the boundary layer stalls. Instead of flow injection, it would also be interesting to experiment suction flows for fan control.

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