Experimental investigation of active control by steady air injection in the tip clearance gap of an axial fan

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Abstract

In an axial fan, a leakage flow driven by the pressure gradient between the pressure side and suction side occurs in the tip clearance gap. This secondary flow is in opposite direction to the working 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 secondary 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 tip clearance 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.

Keywords

Active control — leakage flow — air injection — axial fan — hollow blades

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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

In recent years, more and more investigations on turbomachinery deal with the control by tip injection. In fact, this type control could improve different aspects of the device which could not be enhanced by the geometry refinement. By using injection upstream of the rotor, Weigl et al. [1] succeeded to stabilize rotating stall and surge in a transonic compressor which provided an enlargement of the operating range. Rhee

et al. [2] increased the lift of a hydrofoil by injection control. Eberlinc et al. [3, 4, 5] experimentally increased the pressure of an axial fan composed of 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 [6] experimentally and numerically investigated a transonic turbine cascade with different trailing edge ejection. As Eberlinc et al., the authors successfully reduced the vortex at the trailing edge of the blades.

This paper focuses on the injection used to reduce the phenomenon of leakage flow occurring at the blade tip. This flow driven by the pressure difference between the pressure side and suction side is responsible for important energy dissipations. 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. Neuhaus and Neise [7] attempted to reduce the leakage flow of an axial fan by applying active flow control in the tip region of the impeller. They injected air in the opposite direction of the flow by slit nozzles flush mounted in the inner casing wall. With this process, the authors succeeded to shift the stall point towards lower flow rates and to enlarge the using range of the fan characteristics by 62%. They also increased the pressure of 28% and the efficiency by about 10%. Hamik and Willinger [8] analytically and numerically investigated the concept patented by Auxier [9] in 1995. The idea is to connect 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 is independent of the tip gap width. Hu et al. [10] 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 by a 2D-PIV. As expected, the authors observed a reduction of the leakage flow. However, the tip injection generated a flow phenomenon similar to Karman vortex street in the wake which exacerbate the complexity and non-stationarity of the flow.

In this study, an active control by air injection in the tip clearance gap of an axial fan is experimentally investigated. A new methodology was developed to build hollow fan in which an internal flow could occur. The internal flow, generated by the compressed air system of the laboratory, exited the fan at the tip clearance in purpose to reduce the leakage flow. The results of this control by air injection are presented in this paper.

1. EXPERIMENTAL SETUP

1.1 The axial-flow fan with hollow blades

The fan (Figure 1) used in this study is a controlled vortex designed axial-flow fan with thick and hollow blades [11]. It has six blades with a hub-to-tip radius ratio (R_{int}/R_{max}) equal to 0.337 and a the tip radius (R_{max}) equal to 179 mm. This fan is built in polyethylene by a rotomolding process. A 31-mm axial length and 9-mm thick casing was added into the fan design. It is hollow and circular to increase the fan rigidity. The hollow characteristic of the fan allows a fluid circulation inside. With internal flows into the fan different applications can be developed. In a future work, it is intended investigate the impact of air injection at the trailing edges. According to Eberlinc et al. [3], injecting air at this position increases locally the velocity flow and tends to reduce the boundary layer stalls. It is also planned to experiment suction flows for fan control. In this study, air is injected into the fan and exits at the shroud of the fan. Two configurations are explored: the shroud is composed of 16 or 32 injection holes of 4 mm. For each configuration, the holes are regularly spaced on the shroud of the fan. 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.

The carter used in this study has the same thickness as the fan: 46 mm. Thus, many positions between the fan and the carter are possible. The figure 2 (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 has also a strong influence on the control. The figure 2 (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 ($x_{holes} = 1/2L$).

1.2 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. It is also interesting to determine the aeraulic resistance of the injection circuit in order to quantify injection pressure versus injection rate. A specific drive system shown on Figure 3 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. 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 ±0.2%.

1.3 The test bench

To determine the global performances of the fan, an air suction test bench was used. This experimental facility showed on Figure 4 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 ±0.1 Pa.

2. 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 5. 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 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:

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

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

Figure 1. The hollow fan.

In order to quantify the impact of the control by air injection, the fan characteristics are plotted for 5 different injection rates. They 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 rpm. The black curves represent the fan characteristics with no injection. They 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 [12]. 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 rpm at the design flow rate, $\phi = 0.17$. 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 [7]. As expected, the pressure coefficient increases with the injection rate. Nevertheless, a saturation phenomenon can be observed. For example, on Figure 5(a), an injection rate equal to 3.0% does not seem to affect the pressure coefficient comparatively to an injection rate equal to 2.6%. 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 3.0% of the maximum fan flow rate while at 2000 rpm the ratio is only 1.2%.

To analyze the influence of the number of holes, the figure 6 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 $q_{\text{inj}} = 0$. Each throttle corresponds to a different diaphragm of the test bench (Figure 4) 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 ($q_{\text{inj}}/q_{\text{max}}$) 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 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 spreads across 16 holes seems to better reduce the leakage flow than a lower velocity injection spreads across 32 holes. Regardless the
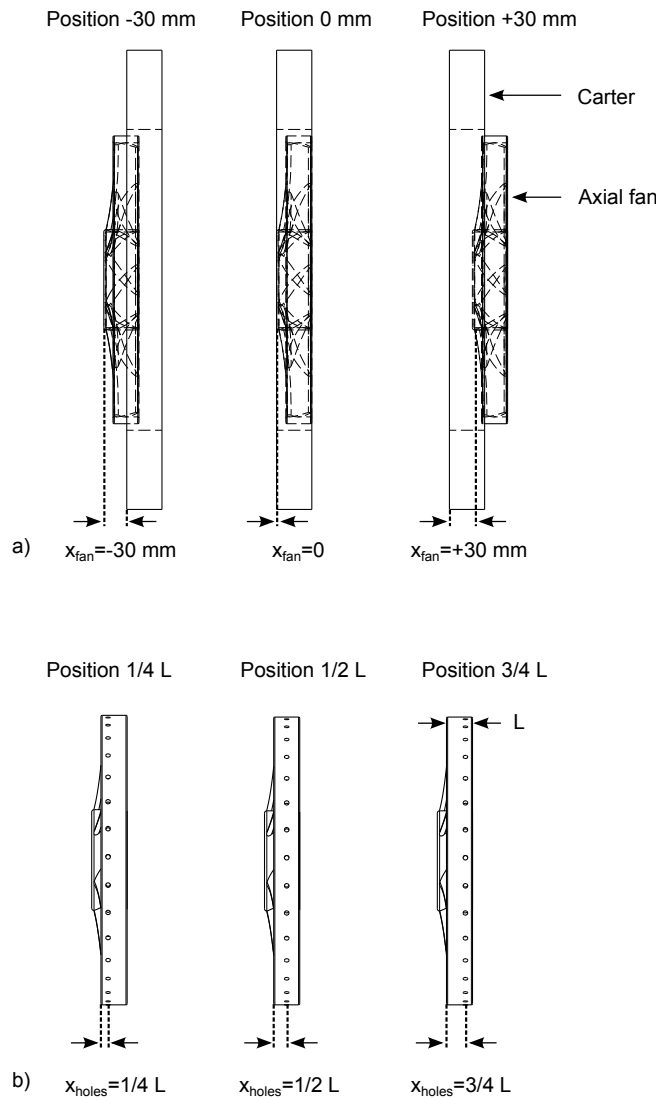
Figure 2. The relative position between the fan and the carter (a) and the position of the injection holes on the casing (b).

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.

3. 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 install a HBM strain gauge transducer to measure torque and extend their investigation on the fan efficiency. As the relative position between the fan and the carter has a strong influence on the active control, it will be investigated in an upcoming study. The authors also 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 be interesting to experiment suction
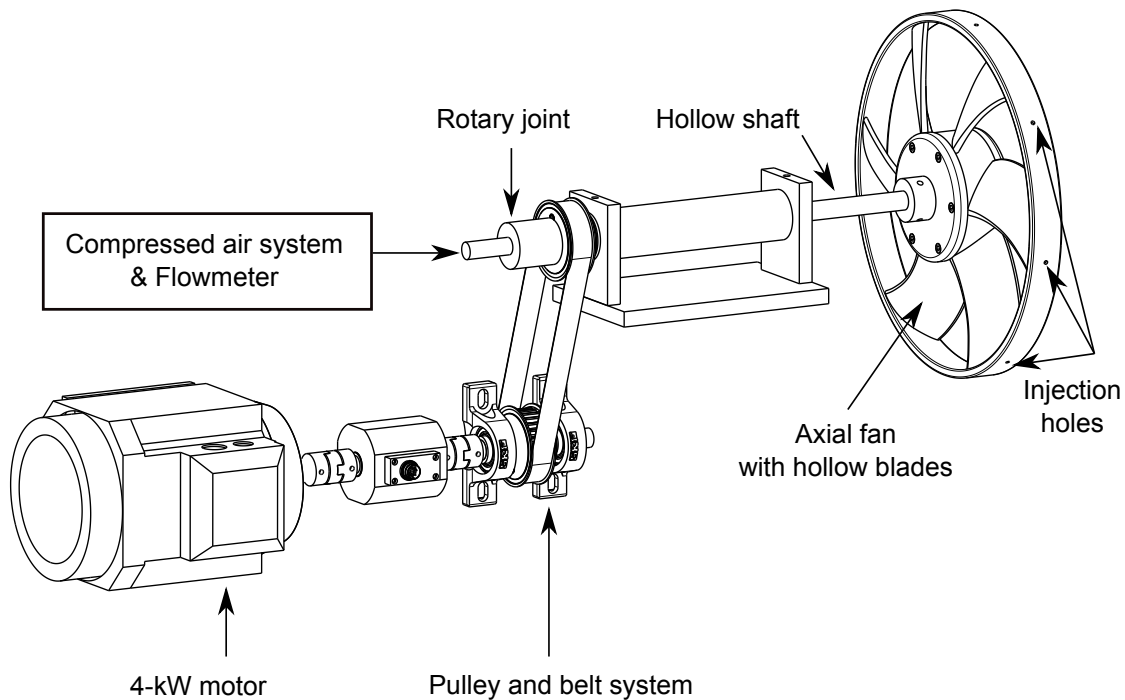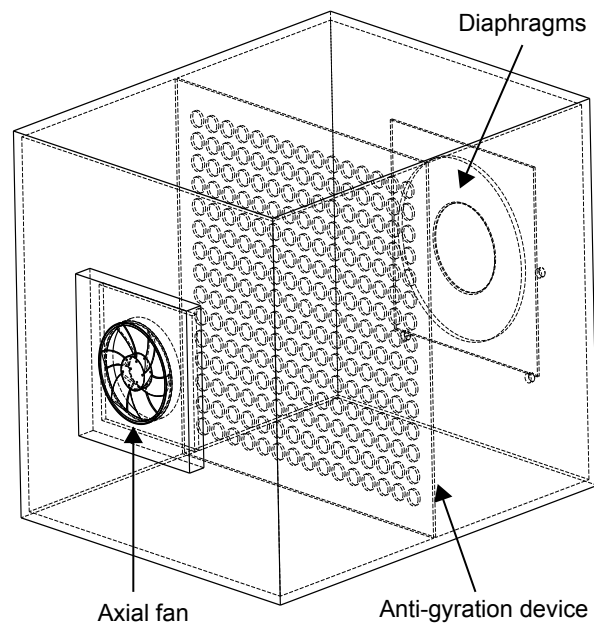
Figure 3. The fan drive system.

Figure 4. 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.

flows for fan control.

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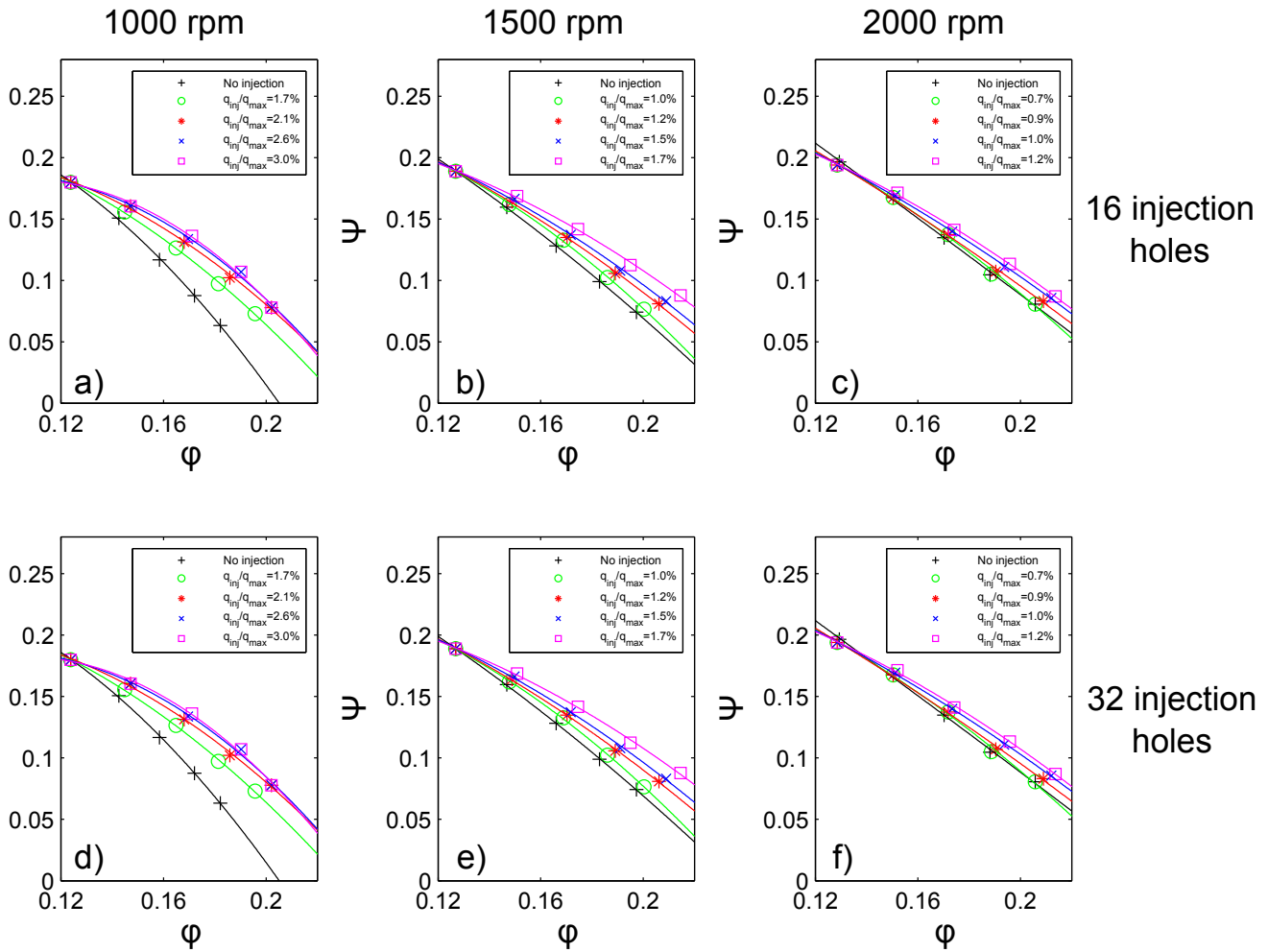
Figure 5. Comparison of aerodynamic characteristic of the axial fan for 5 different injection rates and for 3 rotation speeds: 1000 (a, d) 1500 (b, e) and 2000 (c, f) rpm and for 16 (a, b, c) and 32 holes (d, e, f) at the fan periphery.

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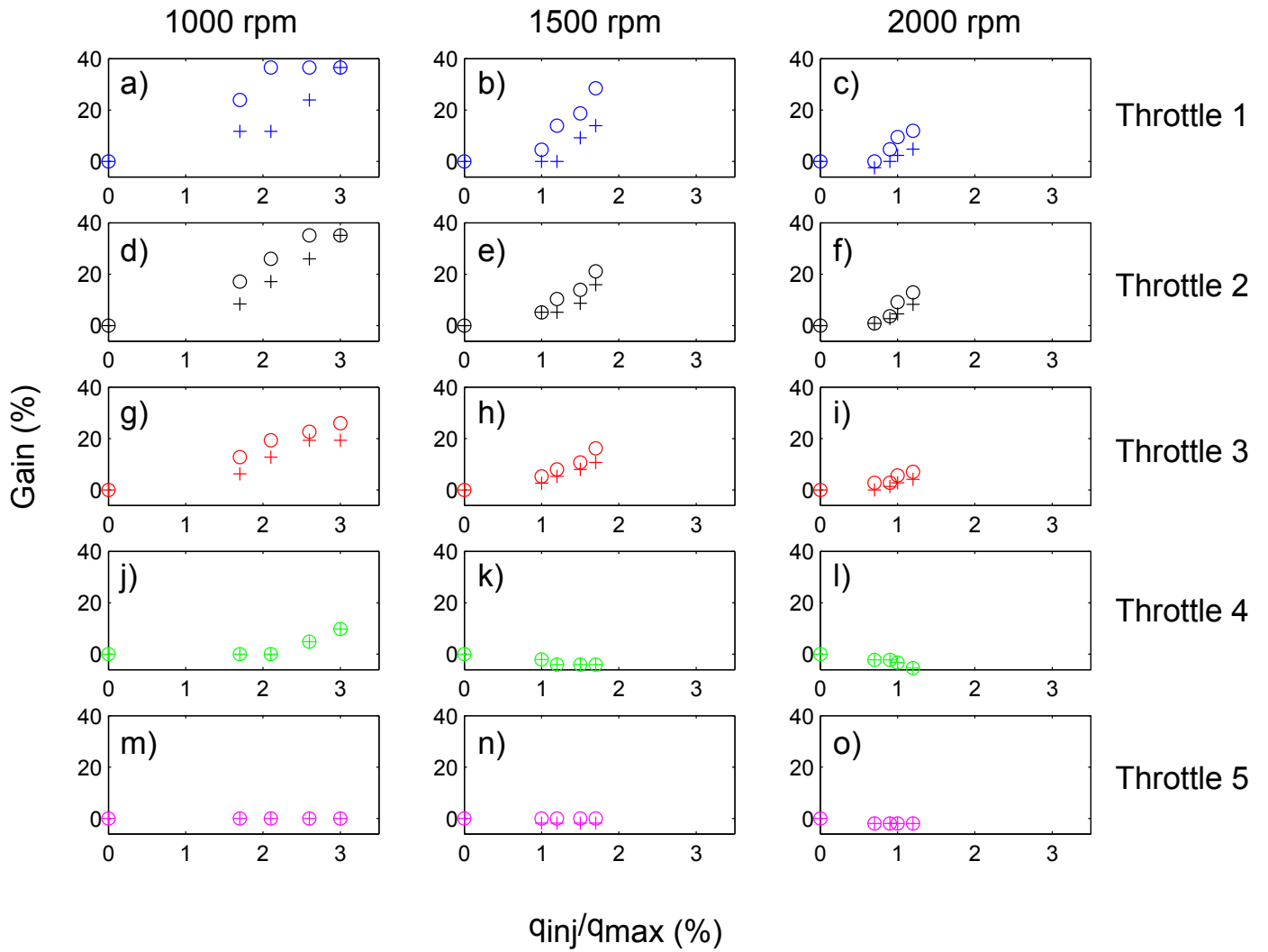
Figure 6. The power gain added by the control for 3 rotation speeds: 1000 (a, d, g, j, m), 1500 (b, e, h, k, n) and 2000 rpm (c, f, i, l, o) and 5 throttles. +: results for 16 injection holes. o: results for 32 injection holes.