IMPACT OF THE NOZZLE GEOMETRY ON THE AEROELASTIC INSTABILITY OF A PLATE SUBJECTED TO AN AIR JET

Antoine Tatin
LadHyX (CNRS-Ecole polytechnique) & Fives Stein, France

Xavier Cluzel, Yannick Mourlot
Fives Stein, Maisons-Alfort, France

Pascal Hémon, Sophie Ramananarivo
LadHyX, CNRS-Ecole Polytechnique, Institut Polytechnique de Paris, Palaiseau, France

ABSTRACT

We study experimentally the flutter instability of a rigid plate supported elastically and impacted by an air jet. The behavior of the plate, modelled as a damped oscillator, is altered by the jet which can generate a negative added damping. As the total damping becomes negative, the plate undergoes growing amplitudes oscillations. We discuss the influence of the geometry of the nozzle, appearing as one of the crucial parameter in the aeroelastic mechanism. The results are compared with the theoretical model of Antoine et al (2008), showing some discrepancies.

1. INTRODUCTION

Gas jets are widely used in industrial processes for cooling or drying due to their thermal properties. This is notably the case of steel strip production lines, where batteries of air jets are used after galvanization, for example, to control the zinc coating thickness and lower the strip temperature. Such steel strips are thin and elongated, which make them particularly flexible and deformable. The interaction with impinging jets can then lead to instabilities, such as self-sustained oscillations or sudden buckling of the strip. The latter are detrimental for the quality of the strips, forcing the industrial line to slow down production.

Jets impinging stationary walls have been extensively characterized (Glauert (1956); Beltaos and Rajaratnam (1973); Fairweather and Hargrave (2002); Maurel et al (2003)), but less is known about their interaction with a moving surface and subsequent instabilities. The few existing studies on the subject investigated fluid-structure interaction mechanisms on simplified model systems, consisting of a rigid plate supported elastically and impacted by normal jets. The motion of the plate is restrained to either pure torsion (Nyirumulinga et al (2008, 2009)) or pure translation (Antoine et al (2007, 2008)), reproducing the fundamental modes of deformation of industrial strips. Instabilities are reported in both cases, which are attributed to the coupling between the plate motion and the flow at the outlet of the jets. Fluid forces can then be expressed as an aeroelastic added stiffness and added damping, which can lead respectively to divergence and flutter when overcoming the structural stiffness and damping of the system.

While only divergence instabilities have been observed for the rotating plate of Nyirumulinga et al (2008, 2009), flutter does occur for the translational set up of Antoine et al (2008) on which we are focusing here. In the latter, a key aspect is the confinement of the flow in the narrow passage between the plate and the jet nozzle, and oscillations develop below a critical plate-nozzle distance. In that respect, the system belongs to the broader framework of flexible plate-like structures subject to leakage flows, which have proved notoriously prone to fluid-elastic instabilities (Mulcahy (1988); Inada & Hayama (1990); Porcher & de Langre (1996); Paidoussis (2004)). The geometry and extent of the confined region is then an important parameter, as confirmed by the results of Antoine et al (2008) on two different planar and circular nozzles. However the impact of geometrical factors such as the jet width and the thickness of the nozzle walls has not been explored yet.

Here, we extend the study of Antoine et al (2008) on the aeroelastic instability of a plate oscillating in translation. We investigate the impact of the nozzle size and wall thickness, and compare the results to the theoretical model of Antoine et al (2008). We further propose potential refinements for the latter to account for discrepancies with experimental results.

2. AN AEROELASTIC INSTABILITY

2.1. Experimental set-up and results

The set-up consists of a 150×90 mm aluminium rigid plate mounted with four identical springs, as illustrated in Fig. 1a. It is subjected to an impinging air jet generated by a nozzle orthogonal to the plate. The latter is connected upstream to a plenum chamber to stabilize the pressure ahead of the nozzle, and a manometer to change the flow velocity. The present set-up does not allow for direct measurement of the velocity at the outlet of the jet U_j. It is instead estimated from the pressure reading of the manometer, using Bernoulli equation. We thus have limited control over its value here, which is dependent upon the pneumatic circuit. As U_j is a key physical parameter, it restricts to a certain extent the interpretation of the experimental results, and on-going work is being done to better control it.

A micrometer screws enable to precisely control the distance H between the nozzle and the plate. Note that the equilibrium plate-nozzle distance slightly increases upon triggering of the jet, and H refers to this shifted position. Finally, the vertical position of the plate is tracked using a laser displacement sensor with an acquisition frequency of 1000 Hz. We extract the angular frequency of the oscillations ω through a fast Fourier transform of the data. Furthermore, the system reduced damping η (explicitly defined later in the text) is measured from the growth rate of the plate oscillations in the unstable case, or its decay rate following a perturbation in the stable case. Namely, local oscillation maximas are detected and the evolution of the logarithmic values with time is fitted with a linear regression. In the paper, each data point for η is averaged over five different acquisitions and error bars corresponds to the standard deviation.

In the present work, we use two different nozzle geometries: a planar and axisymmetric one shown in Fig. 1b. The planar nozzle has an elongated rectangular shape with an inner length $L = 70$ mm that is large compared to the inner width d. The nozzles are 3D-printed, thus allowing to easily change the width within $d = 1-5$ mm, as well as the wall thickness $e = 1-3$ mm that dictates the size of the jet confined region. The axisymmetric nozzle has a circular cross-section with inner diameter $d = 4-6$ mm and wall thickness $e = 1-4$ mm. To change e, we mount 3D-printed collars at the end of the metal tube that forms the nozzle.

Consistently with Antoine et al (2008), we observe self-sustained oscillations below a critical nozzle-plate distance for both geometries. This transition is interpreted in light of the damping in the system η. As reported in Fig.1c-d, we observe that η decreases when reducing distance H, and the critical distance of the instability coincides with η becoming negative. We further compare our results to the theoretical model from Antoine et al (2008), which is succinctly presented in the following.

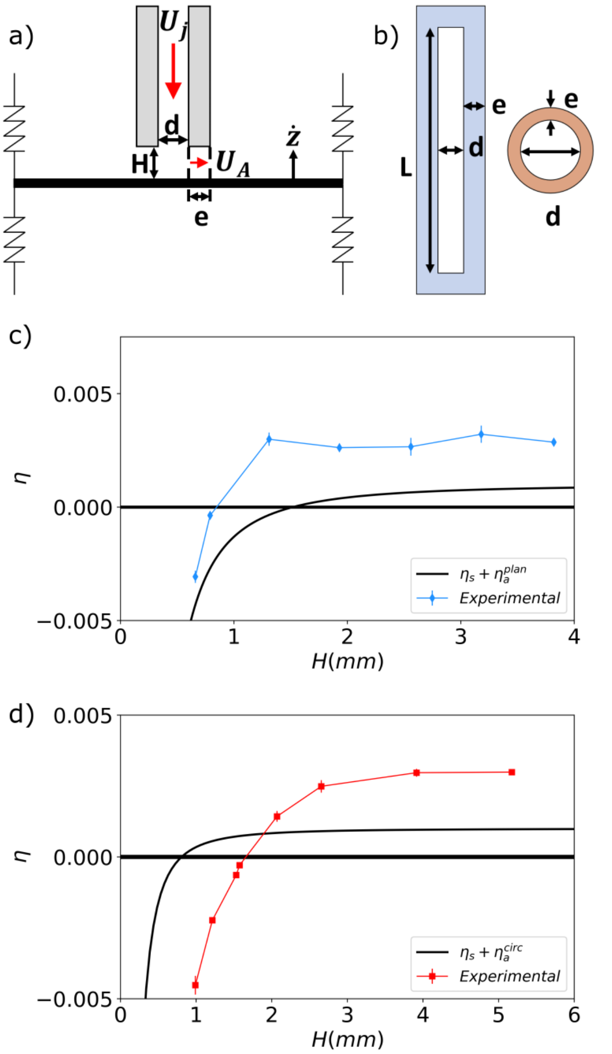

Figure 1. (a) Schematics of the experimental set-up. (b) Nozzles with a planar and axisymmetric cross section. Evolution of the total damping η with the plate-nozzle distance H for, (c) a planar nozzle with $d = 2$ mm and $e = 1.5$ mm, and (d) an axisymmetric one with $d = 4.5$ mm and $e = 0.75$ mm. Experimental results are compared with the prediction of the theoretical model of Antoine et al (2008) (solid lines).

2.2. Theoretical model from Antoine et al (2008)

The plate behaves as an harmonic damped oscillator, and its vertical position z verifies the equation:

$$m\ddot{z} + c\dot{z} + kz = F(z, \dot{z}, \ddot{z}) \qquad (1)$$

with m the mass of the oscillator, k its spring stiffness, and c the structural damping. F denotes the force exerted by the jet as it impacts the plate and seeps through the confined region between the nozzle and the plate. For such leakage flow, pressure is sensitive to the plate motion so that the resulting force F, obtained by integrating the pressure in the confined region, may depend on the plate's position z, velocity $\dot{z}$ and acceleration $\ddot{z}$. It is computed in Antoine et al (2008) based on the model of Porcher & de Langre (1996), in the approximation of small oscillations. The jet force then primarily writes as a negative added damping term η_a, which can negate structural damping and thus trigger the instability:

$$\ddot{z} + 2\omega_0 \eta \dot{z} + \omega_0^2 z = 0 \text{ with } \eta = \eta_s + \eta_a \quad (2)$$

Here, $\omega_0 = \sqrt{k/m} = 77$ s^{-1} is the natural angular frequency and $\eta_s = c/(2m\omega_0) = 8.10^{-4}$ is the structural damping coefficient, both measured experimentally. The expressions of η_a obtained by Antoine et al (2008) for the planar and circular geometry are:

$$\eta_a^{circ} = -\pi\rho U_A \frac{(d/2+e)^3 - (d/2)^3}{3\sqrt{km}} \frac{1}{H} \quad (3a)$$

$$\eta_a^{plan} = -\frac{\rho U_A L e^2}{2\sqrt{km}} \frac{1}{H} \quad (3b)$$

with ρ the air density and U_A the mean flow velocity in the confined region between the plate and nozzle (see Fig.1a). It can be related to the velocity at the jet outlet U_j through arguments of conservation of the flow rate, and we obtain:

$$U_A^{circ} = \frac{d}{4H} U_j \quad (4a)$$

$$U_A^{plan} = \frac{d}{2H} U_j \quad (4b)$$

Note that those expressions for U_A differ from the ones reported in Antoine et al (2008). It modifies the dependency of η_a on the plate-nozzle distance H, which varies as $1/H^2$. The latter is made explicit by expressing Eq.3 as :

$$\eta_a^{circ} = -\pi\rho U_j \frac{(d/2+e)^3 - (d/2)^3}{12\sqrt{km}} \frac{d}{H^2} \quad (5a)$$

$$\eta_a^{plan} = -\frac{\rho U_j L e^2}{4\sqrt{km}} \frac{d}{H^2} \quad (5b)$$

Theoretical prediction for η_a are computed for both nozzle geometries, and the total damping η is compared to experimental results in Fig.1c-d. The model qualitatively captures the observed trends with a sharp decay of η as the plate-nozzle distance is reduced, which eventually becomes negative leading to growing amplitude oscillations. Discrepancies are however observed for the value of the critical distance, which is overestimated for the planar geometry (Fig.1c) and underestimated for the circular one (Fig.1d). In both cases, damping at large H also have lower values than in experiments. However, as will be discussed later, bodies oscillating in a flow are generally subjected to an additional positive damping that is not accounted for in the present model. Adding this contribution would tend to shift the theoretical curve upwards, also affecting the value of the critical distance.

The model captures the decrease of the damping with distance H, but it also indicates a strong influence of the geometry of the nozzle (namely the nozzle width d and wall thickness e) on the onset of the instability. In the following, we study the influence of both geometrical parameters, which were not explored in Antoine et al (2008).

3. INFLUENCE OF THE NOZZLE GEOMETRY

3.1. Influence of the thickness

The wall thickness of the nozzle e is an important parameter as it determines the size of the confined region between the plate and the nozzle. The theoretical model notably predicts a strong dependency of the added damping on e in Eq.5.

We tested planar nozzles with varying thicknesses within $e = 1 - 3$ mm, and the same jet width $d = 2$ mm. Experiments are performed with the same pressure, which is expected to produce similar jet velocities U_j since the nozzles have the same inner cross section. The evolution of the damping η with the nozzle-plate distance H is shown in Fig.2a. Contrary to what was expected, e seems to have little influence on the behavior of the plate and the curves $\eta(H)$ overlap for all the nozzles tested. We also report in Fig.2b the evolution of the angular frequency of the oscillations ω with H, which shows similar results across variations in e. Moreover, we observe that ω increases as the jet is brought closer to the plate, reflecting a rise in the system stiffness. It supports the nature of the underlying mechanism as a dynamic instability, and not a static one (with an added stiffness cancelling the structural one and causing a divergence) as was the case for the rotating plate of Nyirumulinga et al (2009).

We further tested axisymmetric nozzles with different wall thicknesses within $e = 1 - 3.9$ mm and the same inner diameter $d = 6$ mm. As with the planar geometry, we observe little influence of e on the behavior of the plate in Fig.3 for $e \leq 3.4$ mm. However, different results are obtained when the thickness of the nozzle exceeds a particular value close to the nozzle radius $d/2$. Namely, no instability is reported for $e = 3.9$ mm, and contrary to previous results, the damping of the plate tends to increase with smaller H

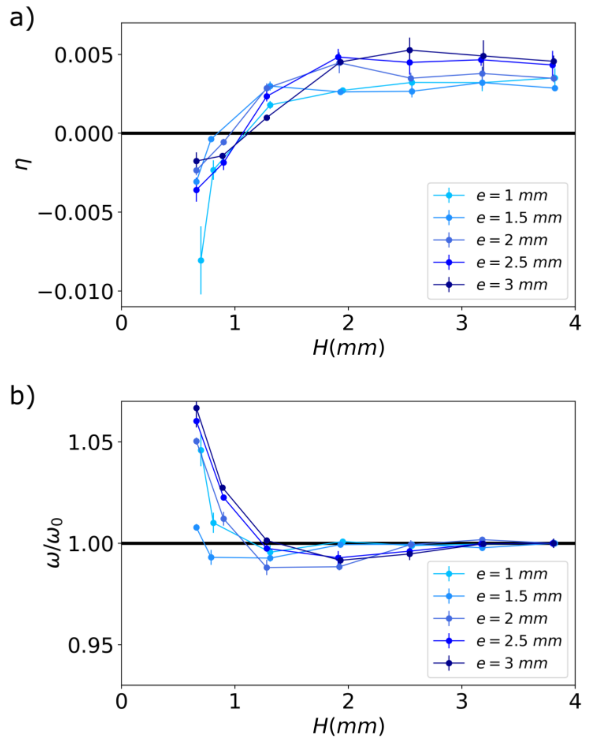

Figure 2. Evolution with H of (a) the total damping coefficient and, (b) the angular frequency of the oscillation ω *(compared to that in the absence of jet* ω_0*), for planar nozzles with different wall thicknesses e and same width* $d = 2$ *mm.*

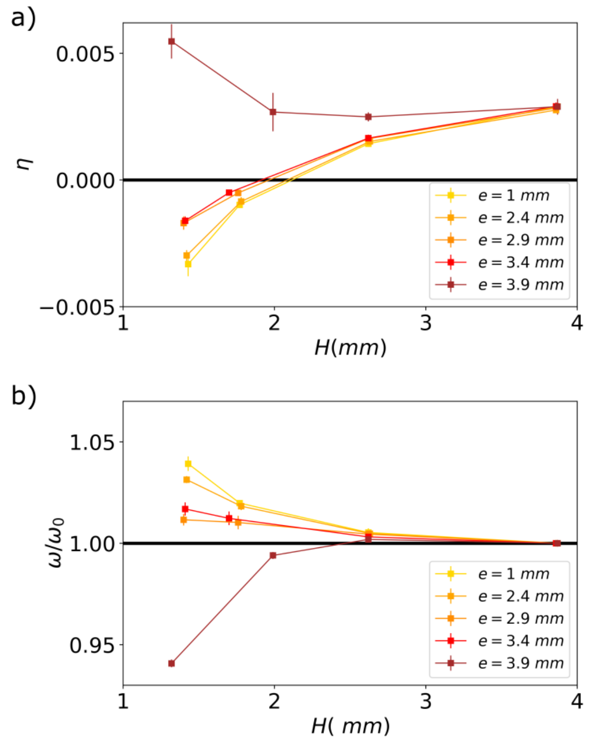

Figure 3. Evolution with H of (a) the total damping coefficient and, (b) the dimensionless angular frequency of the oscillation ω/ω_0*, for circular nozzles with different wall thicknesses e and same width* $d = 6$ *mm.*

in Fig.3a. The evolution of the frequency with H also exhibits a different trend, with a drop of ω reflecting a softening of the system. This phenomenon was also reported by Nyirumulinga (2011). The constriction between the plate and the nozzle's walls results in high-velocity flows, with low pressure according to Bernoulli's principle. For thick nozzles, this low-pressure region is large enough to generate a significant suction force on the plate, which overcomes the pushing force due to the jet impact. It acts in the direction of the displacement and increases as the plate draws nearer to the nozzle, resulting in a negative added stiffness for the system. Further increasing the thickness of the nozzle's walls (with e =5, 10, and 20 mm) amplifies this aerodynamic force, which can then overcome the structural stiffness of the oscillator. In the latter cases, we observe that the plate is jammed close to the nozzle.

3.2. Influence of the jet width

We then study the influence of the jet width d. The latter impacts the ratio between the flow velocity in the confined region U_A and at the jet outlet U_j, as shown in Eq.4. For circular nozzles, varying the jet width also changes the area of the confined region, leading to the dependence on d of the damping coefficient η_a^{circ} in Eq.5a. Results for circular nozzle are however not presented in this paper, due to a large uncertainty on the velocity of the jet. In order to change d, tubes with varying section were adapted at the outlet of the plenum chamber through diverging/converging connectors, and those changes of the pneumatic circuit are suspected to impact U_j. Planar nozzles were directly connected to the plenum chamber, thus reducing those effects.

We tested planar nozzles with different jet width $d = 1 - 3$ mm and the same wall thickness $e = 1.5$ mm. As shown in Fig.4, the total damping of the system is amplified as d decreases, also shifting the onset of the instability towards smaller H. For the smallest width $d = 1$ mm, the plate remains stable (with positive damping) across the range of H explored. While our results show an effect of the jet width, a characterization over a larger range of d is required to identify the dependency of η_a on d, which is the subject of on-going work.

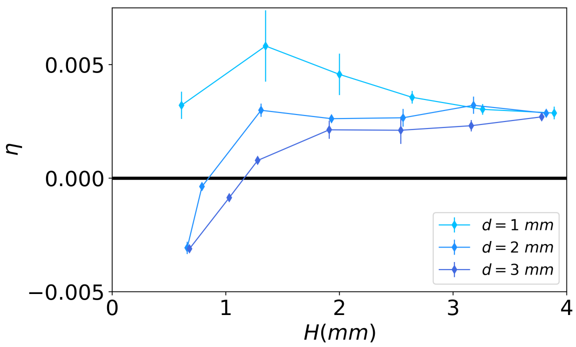

Figure 4. Total damping coefficient as a function of the plate-nozzle distance, for planar nozzles with various jet widths d and same wall thickness $e = 1.5$ mm.

4. POSITIVE ADDED DAMPING

As noted previously, there is a discrepancy between experiments and theory in Fig.1c-d for large distances between the plate and the nozzle. While theoretical predictions for η converge towards the structural damping coefficient η_s for large H, experimental results reach a higher value for both planar and axisymmetric nozzles. The influence of the jet is expected to vanish as we move the nozzle away from the plate (with η decaying down to η_s), but it is still non-negligible for the largest distances tested here of $H/d \sim 4$. In this intermediate regime, the jet may cause an additional positive damping, as is generally the case for objects oscillating in the direction of the flow (Hémon, 2006). The plate motion along the direction of the jet modifies the relative velocity of the incoming flow as $U_J - \dot{z}$. This normal flow exert a force F^* on the plate similar to a drag force, and of the form:

$$F^* = \frac{1}{2}\rho CS(U_J - \dot{z})^2 \tag{6}$$

with C a numerical coefficient to be determined and S the impact surface of the jet. Given the small nozzle-plate distances here, the jet does not have enough space to develop radially and S correspond roughly to the area of cross-sectional area of the nozzle. In the approximation of small oscillations, F^* rewrites to the first order:

$$F^* = \frac{1}{2}\rho CSU_J^2 - \rho CSU_J\dot{z} \tag{7}$$

The zeroth order term, the repulsive force, $\frac{1}{2}CS\rho U_J^2$ is responsible for the initial displacement of the equilibrium position when triggering the jet. The second term depends on $\dot{z}$ and leads to an additional damping term η_j in Eq.2:

$$\eta_J = \frac{\rho CSU_J}{2\sqrt{km}} \tag{8}$$

We estimate the unknown product CS by measuring the static force $\frac{1}{2}CS\rho U_J^2$ for different jet velocities U_J. In practice, we infer it from the displacement of the plate, resulting from the balance between this aerodynamic contribution and the elastic restoring force. We obtains values of $CS^{plan} = 5.10^{-6}m^2$ for the planar nozzle, which corresponds to 3.5% of the jet surface area Ld, and $CS^{circ} = 1.2.10^{-5}m^2$ for the axisymmetric one corresponding to 75% of the jet surface area $\pi d^2/4$. Its yields $\eta_J^{plan} = 4.10^{-5}$ and $\eta_J^{circ} = 2.10^{-4}$. Those values significantly underestimates the gap between experimental and theoretical η in Fig.1, and CS^{plan} is low compared to the surface area of the jet. The measurement of those parameters however depends on the degree of accuracy on U_j, which is limited with the present set-up. Additionally, the estimate can be improved by taking into account the velocity gradient in the direction of the jet (Hémon (2006)).

5. CONCLUSION

In line with Antoine et al (2008), we show that a plate subjected to an air jet undergoes a flutter instability, which is attributed to a negative aeroelastic damping generated by the flow going through the narrow passage between the plate and the nozzle. We further studied the influence of the nozzle geometry, and notably the thickness of the nozzle's walls which dictates the extent of the confined region. Contrary to what was expected and predicted by the model of Antoine et al (2008), it seems to have little influence on the behavior of the plate. In contrast, changing the nozzle inner width significantly modifies the critical distance below which oscillations start to grow. While the model captures the effect of the width, it is not consistent with the results for different wall thicknesses. Ongoing work is pursued on refining the model to account for it, but also to extend experiments. In particular, a limiting factor of the present set-up is the lack of direct velocity measurement at the jet outlet. A sensor will be mounted on the nozzle to control it more precisely. It will notably allow to evaluate the additional positive damping of the jet observed at larger nozzle-plate distances.

6. ACKNOWLEDGEMENTS

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