Synthetic Jet Cooling Part II: Experimental Results of an Acoustic Dipole Cooler

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Abstract

The paper discusses experimental results for a typical embodiment of synthetic jet cooling technology: an acoustic dipole cooler comprised of a standard loudspeaker in a housing provided with two pipes. A transient measurement set up is used to measure the average heat transfer coefficient based on cooling a 5*5 cm^2 metal plate. Heat transfer and noise results are presented for a range of frequencies, pipe lengths and diameters. The results are compared with a standard 60*60 mm fan. It is concluded that, at least for the cases studied, the synthetic jet is superior on all fronts: heat transfer performance, noise level and dissipated power.

1. Introduction

A short overview of the literature related to the principles of heat transfer and acoustics is presented in an accompanying paper[1] that also discusses the benefits over fan-driven forced convection and the challenges related to numerical analysis. The conclusions in short: significant improvements can be expected in terms of heat transfer, acoustics, reliability and design freedom. Unfortunately, due to the very complex fluid dynamics, the current state-of-the-art CFD codes are not suited for design purposes. A promising way to circumvent the numerical problems associated with synthetic jets seems to adapt the theory of continuous jets. The present paper starts with a description of the experimental set up for measuring the heat transfer performance, followed by a sketch of the dipole cooler. The remaining of the paper is devoted to a discussion of the experimental results for a range of pipe dimensions and frequencies. A comparison is made with a standard fan.

2. Set up for the measurement of the heat transfer coefficient

One of the options to compare the cooling performance of various cooling methods is to measure the average heat transfer coefficient over a certain area. Due to the differences in air flow generation between synthetic jets and fans it is far from trivial to compare the two cooling methods in a general way. Currently, discussions are ongoing with Georgia Tech to reach consensus on a proposal for standardization and/or guidelines of comparison methods. For the moment, we decided to measure the average heat transfer coefficient over an arbitrarily chosen area of size 5*5 cm^2 by using a transient method instead of the more common steady state methods. The advantage is that no heater is required on the plate to be cooled, only a measurement of the temperature as a function of time after heating the plate on a hot plate. Corrections dealing with heat transfer by radiation and conduction losses via the substrate are accounted for in a curve-fitting program. Figure 1 shows the basic layout.

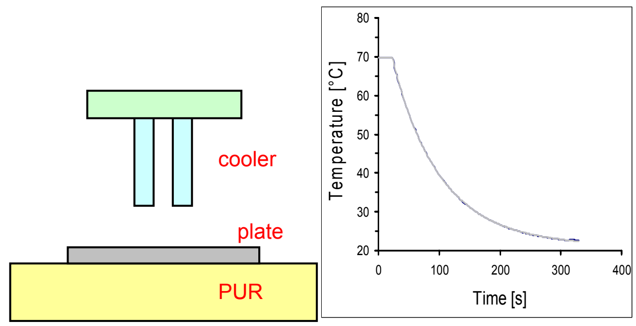

Figure 1: Sketch of test set up and typical cooling curve

Synthetic jet design used in the test set up

Figure 2 shows a typical layout of a so-called dipole cooler showing the two pipes attached to both sides of a single loudspeaker with a diameter of 25 mm. The volumes V1 and V2 (size about ~ 2 cm^3) at both sides should be equal, which is realised by a screw that enables an accurate matching of V1 and V2. Measurements have been performed with pipes with inner diameters of 3, 4 and 6 mm and lengths of 30, 60, 90 and 120 mm, for various frequencies and voltages.

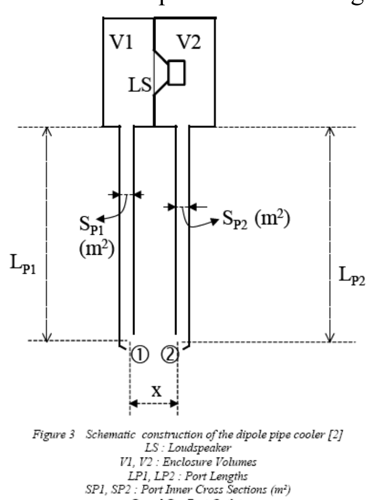

Figure 2: Sketch of dipole cooler

3. Experimental results: heat transfer

Preliminary experiments showed that the optimal distance between pipe exit and plate was about 4 cm, see for a typical result Figure 3. It is clear that the optimum around 4 cm is not very sensitive to the exact distance.

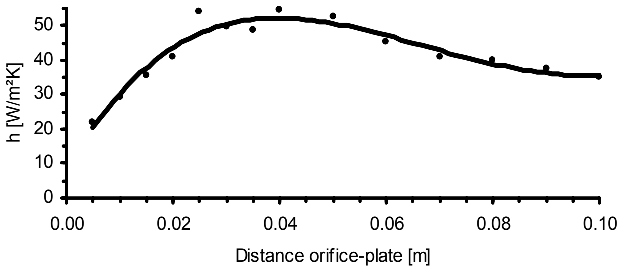

Figure 3: Heat transfer coefficient h vs. distance between pipe exit and plate

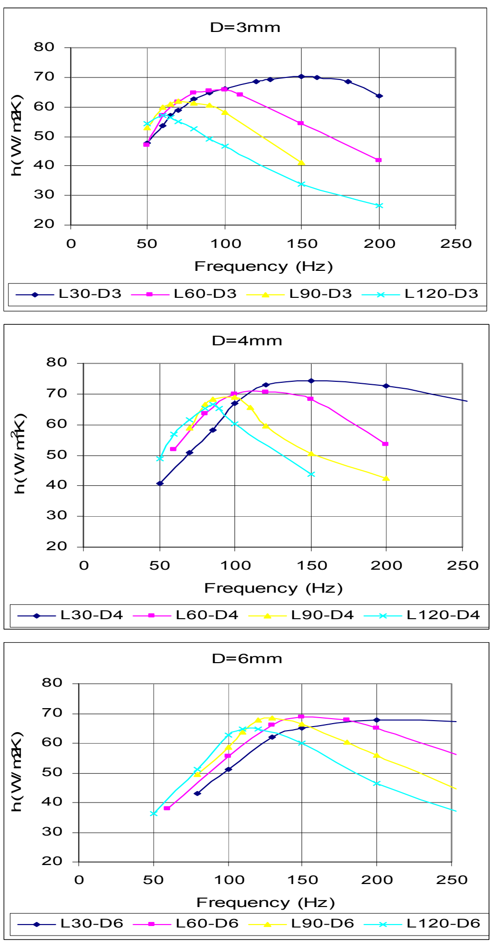

Figure 4: Heat transfer coefficient h as a function of frequency for all diameter-length combinations

Figure 4 shows the average heat transfer coefficients measured on the 5*5 cm^2 plate as a function of the frequency for all diameter-length combinations, for a fixed distance pipe-plate of 4 cm and a voltage applied to the loudspeaker of 3 V. As is clear from the graphs, maximum heat transfer coefficients are between 60 and 75 W/m^{2}K. This should be compared with a measured value of about 9 W/m^{2}K for natural convection/radiation for the horizontal blackened plate. Closer observation of the graphs reveals that a maximum exists for every diameter-length combination at a certain frequency. This frequency shifts to higher values when the pipe length is decreased whilst maintaining a fixed diameter, or when the diameter is increased for a fixed pipe length. From a noise level point of view the frequency should be as low as possible. In other words, the best combination is the largest pipe length and the smallest diameter. Unfortunately, the maximum heat transfer tends to decrease with lower frequencies. Uncertainty based on replication and repetition is estimated to be of the order of 5% or less.

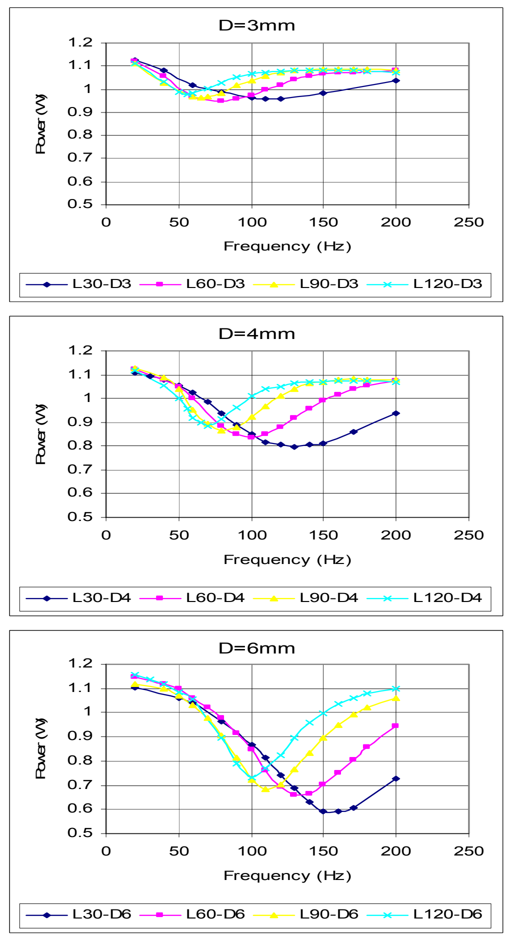

Figure 5: Driving power P as a function of frequency for all diameter-length combinations discussed in this paper

Figure 5 shows the power that is needed to drive the dipole as a function of frequency. It is clear that the same trends are

observed. Also, the larger the diameter, the higher the frequency at minimum power but also the lower the required power. It is interesting to compare the relatively difficult to measure heat transfer coefficient with a few much more easy to measure parameters. One is the already discussed power, two others are the centreline velocity and the impulse. The velocity measurements were performed with an ATVS2020 tester using small thermistor beads instead of a hot wire, rendering the set up much more robust at the cost of frequency sensitivity. The maximum centreline velocity at a distance of 4 cm was recorded.
For the impulse measurements we used a simple balance, again with the exit of the pipes at a distance of 4 cm.

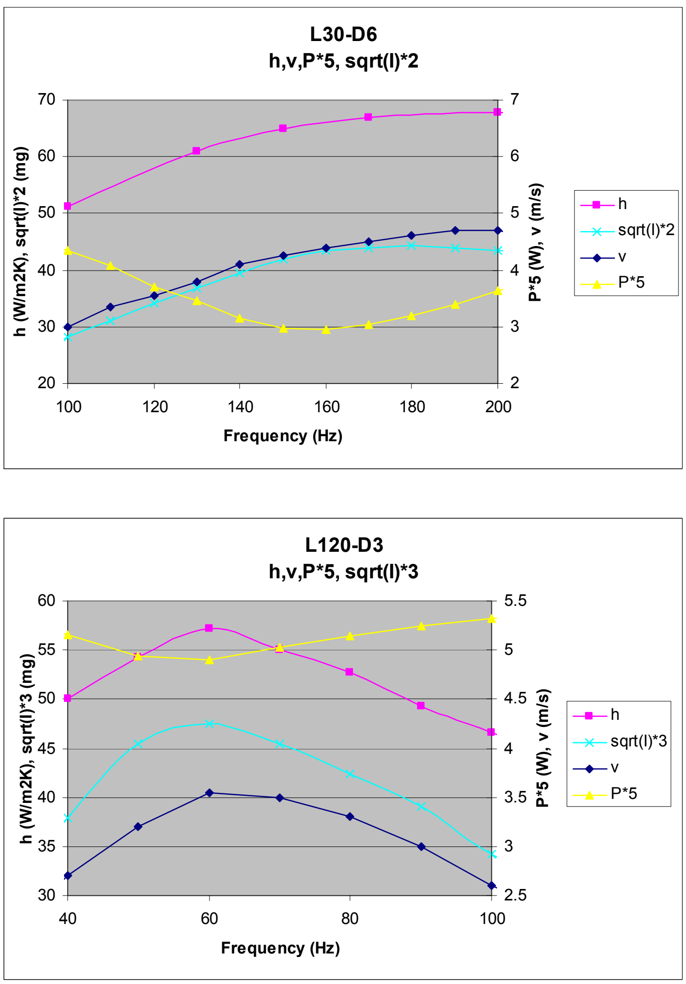

Figure 6: Plots of heat transfer coefficient h, power P, velocity v and square root of impulse I : L30-D6 and L120-D3

Figure 6 plots for two different D-L combinations. It is clear that the maxima in heat transfer are more or less co-located with the minima in driving power, albeit with a little shift towards higher frequencies for h. The velocity and h measurements are congruent, as expected. The same conclusion is valid for the square root of the impulse, providing a means to extract the average velocity over the cross-section of the jet. Please not the scaling factors to plot all parameters on the same graph. The extraction of design rules for jet impingement situations correlating the single-point velocity measurement with the heat transfer coefficient given the area is the subject of further study.

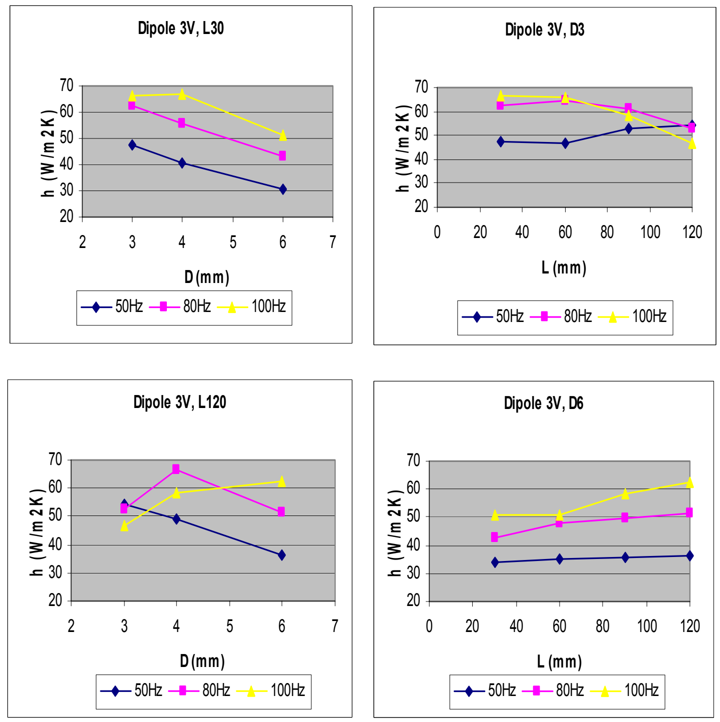

Figure 7: Plots of heat transfer coefficient h as a function of D and L for three frequencies

Another important design parameter is the dependence of the heat transfer coefficient on the diameter and length of the pipe, given a certain actuator. Figure 7 shows clearly that we are dealing with severe nonlinearities. Concentrating on the lower frequencies, there seems to be an optimum around D=4mm, while the influence of the length is less pronounced. Of course, the length does influence the frequency for minimum power, and as such the driving power.

4. Acoustical results

Figure shows a typical example of the simulations of a certain dipole configuration, in this case for D=6 mm, with L as the parameter. The simulations are based on a relatively simple linear lumped-element acoustic model, providing information on SPL, displacement, exit velocity and impedance. Because of the symmetry of the dipole, only the output of one port is shown.

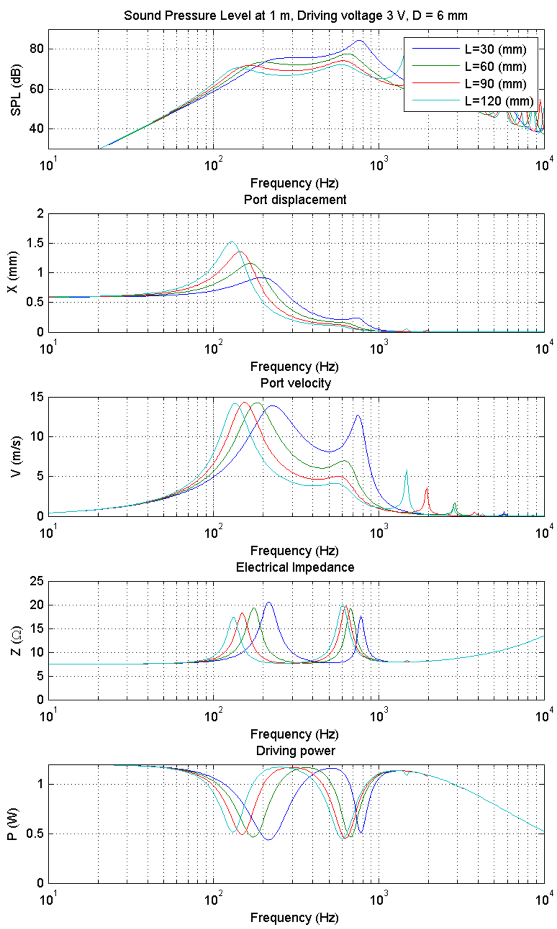

Figure 8: Simulations of a single port of the dipole cooler, D=6 mm with pipe length L as the parameter

The model to generate Figure 8 contain lumped volumes V_1 and V_2 (see Figure 2) coupled to a transmission line model for the pipes[2], where the acoustical damping in the pipes is considered to be fixed and frequency independent. A short summary is given in the accompanying paper[1]. If we approximate this system by a simple Helmholtz resonator with resonance frequency f_H according to

$$f_H = \frac{c}{2\pi}\sqrt{\frac{S_p}{L_p V_1}}$$

where S_p is the pipe area, V_1 the volume (e.g. as shown in Figure 2) and L_p the pipe length, then it follows immediately that for a fixed volume the Helmholtz frequency follows

$$f_H \propto \frac{D_p}{\sqrt{L_p}}$$

In summary, the first-order theory predicts that the frequency that is associated with the minimum in the power scales with the diameter D and is inversionally proportional to the root of the pipe length L. This relation is fairly well supported by the experiments, over the whole range of D between 3 and 6 mm and L between 1 and 1000 mm, as is clear from Figure 9.

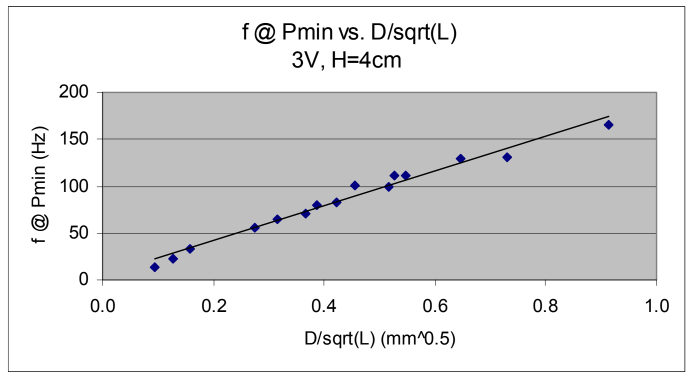

Figure 9: Dependence of frequency @ minimum power on pipe diameter and length

Special attention is asked for the velocity and power plots. A bird's eye comparison with the graphs of Figure 4 and Figure 5 clearly shows the same trends in the location of the maxima and the dependence on the length of the pipes. However, looking in more detail, some discrepancies become obvious. For example, while the magnitude of the minima in the power curve is correctly predicted, the frequency is shifted to the right by about 15%. Also, the calculated maximum centreline velocity is over predicted by a factor of three. These differences are all caused by the assumption of linearity of the model. Only viscous losses are taken into account, turbulent losses are neglected. To check this influence in practice, the voltage was reduced from 3V to 0.3V, effectively decreasing the power by a factor of 100. For the L30D6 pipe, the resonance frequency shifted from 170 Hz to 205 Hz, close to the calculated value of 210 Hz. Further to this, it appeared that including nonlinear losses in the model, the calculated maximum velocities were reduced to realistic values.

In addition to the calculated SPL, the sound power levels were measured in a reverberant chamber according to International Standard ISO 3741:1999.

For comparison with conventional technology a quality fan of size 60*60 mm was selected. Table 1 shows the results for some selected pipes and the fan at two different voltages, tested under the same conditions.

	f	h	exit area	SPL	noise	driving power
	Hz	W/m^2K	mm^2	dB	dBA	W
D3-L30	110	68.7	5	43.1	38.2	0.96
D3-L90	100	56.7	5	41.5	29.7	1.04
D3-L120	65	55.1	5	36.9	27.2	0.99
D4-L30	110	65.5	25	46.4	40.4	0.82
D4-L90	80	66.7	25	48.1	38.2	0.87
D4-L120	65	58.3	25	49.0	37.3	0.90
fan, 6V	n.a.	39.7	2500	36.6	35.2	0.5
fan, 12 V	n.a.	58.4	2500	49.5	48.5	2.0

Table 1 : Thermal performance for several cooling options, driving voltage 3V

When comparing both fan results with for example D3L120, it is clear that the synthetic jet outperforms the fans, at least, for the selected area of 5*5 cm^2. To get a feeling about the dependence on the area, a series of experiments was carried out for a range of areas varying from 2*2 to 10*10 cm^2. The question is: can we predict these trends? There is not much information in literature that is useful from an engineering point of view. All correlations published for synthetic jets use dimensionless numbers that are related to the stroke length which itself is a function of a time and area-averaged velocity scale, for which determination sophisticated measurements are required. This is in contrast with continuous jet theory where in many cases only the maximum centreline velocity is needed, e.g. Martin[3]. His equations are used to compare the area dependence as measured for a number of coolers with continuous jets for the same geometrical parameters. Unfortunately, the problem with the dipole is that it is rather difficult to allocate a number for the effective diameter to compare its performance with other synthetic jets that consist of only one pipe. In one particular experiment only one pipe of the dipole was used to get the average heat transfer coefficient, resulting in values that are about 20-25% lower than the values for the two pipes which indicates that the distance between the pipes is not optimized. Of course, matters are much more complicated because also the distance between the pipes, the diameters, the height and the area are dependent parameters. Despite all these theoretical considerations that certainly deserve academic attention we need engineering information on a shorter timescale. Figure 10 gives such information. Two different coolers with one pipe, the dipole and the fan are compared with continuous jet theory, assuming an exit velocity of 6.5 m/s, for two different pipe diameters. The correspondence between the various curves is striking. It can be concluded that the synthetic jets, with their exit area of about 20 mm^2, perform better for areas below about 40cm^2, as compared to the fan with its more than hundredfold larger exit area.

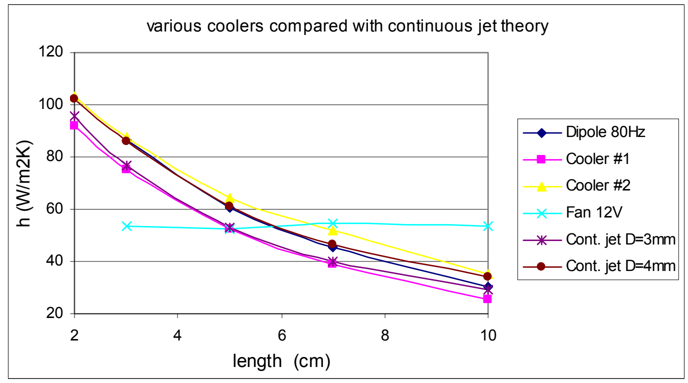

Figure 1: Heat transfer coefficient as a function of cooled area for various coolers

5. Further research

Despite the promising results, many questions still need to be answered due to the complex nature of the underlying physics. The question is: can we extract design rules that are both simple and make sense, linking important parameters such as the heat transfer coefficient, the pipe diameter and length,, the frequency, the driving power and the noise? Let's again have a look at Figure 7, the one showing the dependence of h on both length and diameter. It is obvious that a correlation that captures all trends will be very complex and hence cannot serve as a guideline.

Building knowledge is instrumental for the ultimate goal: providing the designer with a method for creating the most optimal cooling solution for the product at hand. The problem is really the abundance of design parameters and the fact that this requires a fundamental understanding of the coupling of acoustics and heat transfer, a field that is largely unexplored. In addition, it should be realised that synthetic jets and fans are difficult to compare, and it might well turn out that both cooling technologies will have their own application fields. In the future, we intend to publish about other promising embodiments reducing the noise level even further. On the other hand, it should also be realized that the technique allows for much more design freedom than fans, that its much smaller exit area has pros and cons, and in particular, that its implementation is not just about replacing fans by jets. The maximum benefits will be acquired by completely rethinking the thermal management concept.

Finally, work needs to be done in the areas of standardization and of compact model generation to enable designers a fair comparison with other cooling technologies.

Conclusions

Heat transfer and acoustic experiments were performed for an acoustic dipole consisting of a small off-the-shelf loudspeaker of which both sides are connected to pipes of various diameters and lengths, as a function of frequency and driving voltage. Actuating the loudspeaker causes synthetic jet formation at the end of both pipes. Heat transfer was measured over a certain area cooled by normal jet impingement created by the synthetic jets. Measurements of the heat transfer coefficient, the centreline velocity and the impulse were compared with a linear lumped-element model showing good agreement in all trends observed.

Acoustic measurements were done in an anechoic chamber and a reverberant room. The results were compared to a standard fan.

The results show both the complexity and the potential of this relatively novel cooling technology. Below an area of about 40 cm^2 the synthetic jet outperforms the fan, at least for normal impingement, both from a heat transfer and an acoustic point of view. A very important design objective is to lower the frequency for which the driving power is minimal. Many parameters play a role in achieving this goal, and simple design rules are not to be expected for some time.

Finally, the real challenge is in embedding this technology in practice, because it is not just about replacing fans by synthetic jets.

Acknowledgments

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