Synthetic Jet Cooling Using Asymmetric Acoustic Dipoles

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Abstract

In two earlier papers [1, 2] the principles and experimental results have been discussed 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. The current paper shows experimental and numerical results for another type: the asymmetric dipole. Basically, this type consists of a loudspeaker with a minimal volume attached to it with one or more holes with or without pipes. Results for driving power and noise are presented for a number of actuators covering a large parameter space: frequency, pipe dimensions and driving voltage were varied over a large range. A relatively simple acoustic model extended to include separation losses matched the experimental results very well. The results indicate promising heat transfer performance with minimal noise combined with a large degree of freedom.

1. Introduction

A short overview of the literature related to the principles of heat transfer and acoustics is presented in [1] that also discusses the benefits over fan-driven forced convection and the challenges related to numerical analysis. A second paper [1] showed experimental results for a dipole cooler. Summarising the paper: heat transfer was measured over a certain area cooled by normal jet impingement. Measurements of the heat transfer coefficient, the centreline velocity and the momentum flux 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 typical 60*60 mm fan (due to historical reasons blowers were not taken into account in the comparison but should be the subject of further research). The results showed 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. In the conclusions it was stressed that a very important design objective is to lower the resonance frequency for which the driving power is minimal. This objective was the main driver for the continuation of our research. The paper deals with the following additional topics: extensive results for various single jet coolers, e.g. heat transfer, noise, velocity, momentum flux, for a wide range of frequencies and pipe dimensions, a draft version of a standardisation proposal, new insights in the 'history' of synthetic jet cooling, and proposals for further research. Additionally, subsections cover results for long pipes (up to 2 m) and some preliminary measurements with sealed enclosures.

2. Relevance

A cooling technology that could significantly enhance natural convection while at the same time relaxing many of the drawbacks of a fan would be an interesting option for many electronic products. One of the most promising options is to use jets, in particular synthetic jets. Basically, the technology requires only a small loudspeaker and some electronics.

The benefits are somewhat dependent on the application, but in general the following improvements are expected when comparing synthetic jet cooling with fans, for the same heat transfer performance:

- lower noise level
- better (thermodynamic) efficiency, half the power or less
- design-friendly due to potential separation of nozzle and actuator
- intrinsic higher reliability
- less fouling problems, the vibrating part can be protected from the ambient
- miniaturisation easier than with fans (sub mm possible)
- relatively simple noise cancellation possible
- remote cooling feasible.

To be complete, we should also mention the disadvantages:

- fans provide (much) more air for the same power (not the same as heat transfer performance)
- large-area cooling with few jets may be problematic
- prevention of backflow of heated air is not trivial
- amplitude and size of actuator are related, prohibiting very efficient thin designs. However, this might be relaxed by enabling larger actuator areas
- fans are commercially available on a large scale.

3. Physical principles of synthetic jets

For a more detailed discussion the reader is pointed to [1, 2]. Figure 1 shows the most important features:

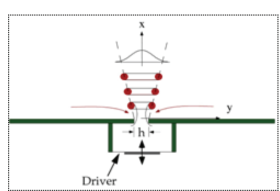

Figure 1 Sketch of synthetic jet formed by an actuator in a cavity provided with an orifice

In more technical terms, it is a device that requires zero mass input yet produces non-zero momentum output. They are synthesized from the coalescence of a train of vortices as a result of a time-harmonic motion of a diaphragm driven by whatever means, piezoelectric, mechanical or magnetic, provided the amplitude is large enough to induce flow separation at the orifice.

4. The asymmetric dipole

We discuss a new technique complementary to the existing ones. It is obtained by introducing an (almost) equal mass flow at the orifice of a small pipe and-apart from a minus sign-at the back of the loudspeaker. Acoustic waves from both ends may interfere resulting in the dipole effect. The result is a dipole radiator while we do not need a mechanical symmetrical construction as was the case for the dipole coolers discussed before. Figure 2 shows a general dipole where V0 may also be 'infinite', hence representing the environment. The waves coming from the backside of the loudspeaker directly or via large holes in the cavity V0 will interfere with the waves coming from the orifice of the tube, and will cancel some of the unwanted noise. Because of the asymmetric construction we refer to the device as an 'asymmetric dipole'.

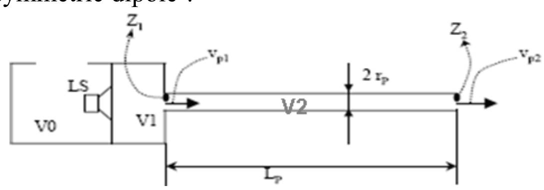

Figure 2: General picture of a dipole showing volumes V0, V1 and V2, with loudspeaker LS

We tested two types of loudspeakers, a 50 mm Philips PSS41 and a 30 mm LS Veco 32KC08 and compared them with the fan already discussed, the dipole coolers and a typical commercially available synthetic jet actuator of 50 mm diameter.

4.1. Results for the PSS41

This section provides the most important results for the 50 mm PSS41 loudspeaker. The main objective of the experiments was to gain insight in the relations between performance criteria such as the heat transfer coefficient, the dissipated power and the noise and controllable parameters such as pipe dimensions, frequency and voltage. Additional results based on velocity and momentum flux measurements are available but are not shown because they are related in one way or another to h (see also [2]). Figure 3 shows some results for two pipe lengths (30 and 60 mm) and four diameters: 3-4-5 and 6 mm, at a driving voltage of 3 V.

It is clear that selecting the right length-diameter combination can significantly reduce the power required. Maxima in h up to about 80 W/m^2K are to be found at higher frequencies.

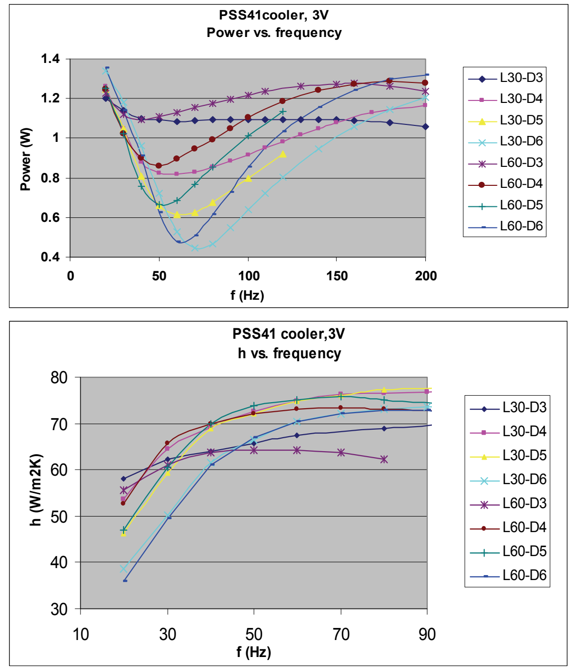

Figure 3: Top: electrical power vs. frequency for various length-diameter combinations, Bottom: h vs. frequency

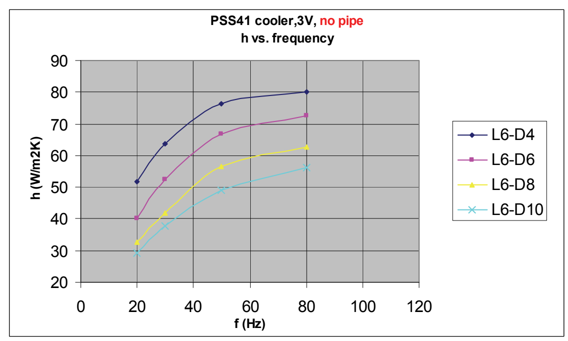

Figure 4: h vs. frequency for 'zero' length pipes

In Figure 4 some results with no pipe attached to the hole in the cover plate. 'L6' corresponds to the thickness of the cover plate of the jet embodiment (6mm). Note that the graph shows a promising regime at lower frequencies where h is still relatively high. For example, at only 20 Hz we still measure values up to about 60 W/m^2K.

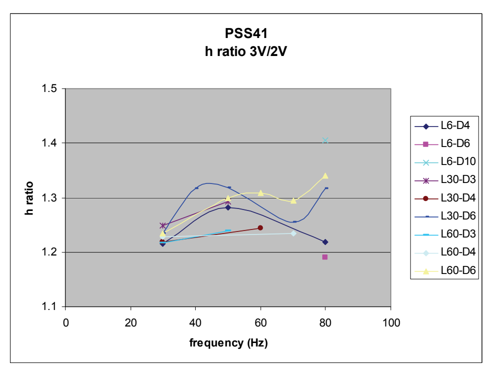

Figure 5: Ratio 3V/2V for h vs. frequency

Figure 5 plots the ratio between the results for 3V and 2V. If linear behaviour is assumed, going from 2 to 3 V should more than double the power, and hence the momentum flux and the velocity squared. Because h scales roughly with v to the power 0.8, the h ratio should be about 1.4. This is clearly not the case for any combination. Around 50 Hz the maximum ratio is about 1.3 for a 6 mm diameter pipe.

In fact, the conclusion is that by doubling the power h increases roughly by something between 20 and 30%.

4.2. Slit orifices

In a number of applications it might be of interest to keep the exit area because of performance reasons but to 'squeeze' the round pipe in such a way that the area is retained, resulting in a slit-like essentially two-dimensional exit.

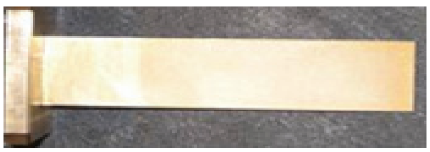

Figure 1 Picture of the 1*28 mm slit

The question is: at what slit width do we see deviations from the round jet performance due to boundary layer effects causing the effective exit area to be smaller? To get an impression, four exit areas of the same area were taken: round 6 mm, slit 4*7, 2*14 and 1*28 mm. From Figure 7 it is clear that not much is changing The reason is thinning of the boundary layer due to the pulsing nature of the jet.

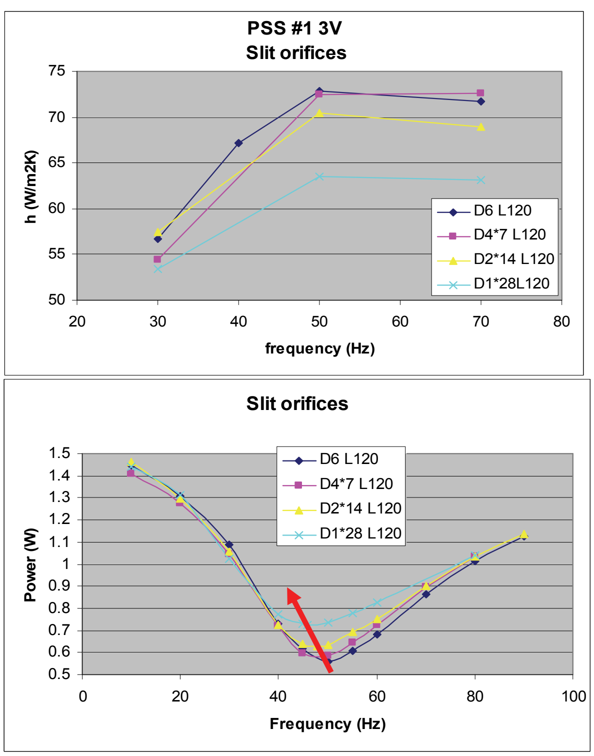

Figure 7: h and power versus frequency for various exit area shapes

From the results it is possible to get a first order estimation of the boundary layer thickness, as follows. We know from basic acoustic theory that the resonance frequency scales linearly with the pipe diameter. Figure 7b shows that f @ P_{min} shifts to the left with smaller area/perimeter ratio due to a smaller *effective* area caused by boundary layer effects. When we equate the effective area of the slit with width a and height b corrected for the boundary layer thickness δ to an equivalent round pipe area with diameter D_{eff}, we get:

$$(a-2\delta)(b-2\delta)=\pi/4D_{eff}^{2} \quad (1)$$

D_{eff} is estimated from the graph that plots f@P_{min} versus pipe diameter, obtained from the already known data for the three pipes with diameter 3, 4 and 6 mm, as shown in Figure 8. The f@P_{min} data acquired from Figure 7 are used in the regression equation shown in Figure 8 to obtain D_{eff}. Solving for δ using Eq.(1) results in the values we are after.

For all configurations tested, the boundary layer thickness is of the order of 60 µm.

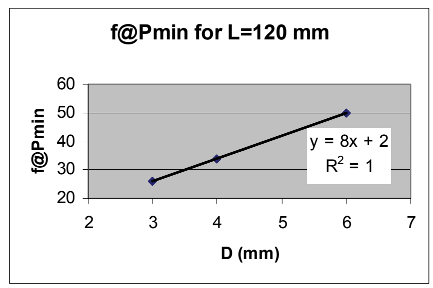

Figure 8: Graph of f@Pmin as a function of pipe diameter

5. Comparison of various coolers

Below h and driving power data are presented for the Veco and dedicated synthetic jet actuators, these should be compared to the data shown in Figure 3. It is clear that the PSS41 outperforms the other coolers in terms of low frequency and high h.

Clearly some trends are visible, and relevant conclusions can be drawn for the typical actuators investigated. The ultimate aim is to generalise the results in order to enable the prediction of the behaviour of other actuators. For the electrical and acoustical part we made some significant progress in this sense, as is presented at the end of the following section.

6. Long pipes

Many application fields exist where long pipes are already part of the system, especially in the lighting business. As explained in [1] synthetic jets exploiting long pipes are more related to transmission lines than Helmholtz resonators. Synthetic jets exploiting long pipes (we tested the range from 100 to 2000 mm) show a very different fluid flow compared to fans and blowers. For high enough frequencies (say >100 Hz) little air is transported through the pipe, all flow is generated at the exit area. For low frequencies the physics are

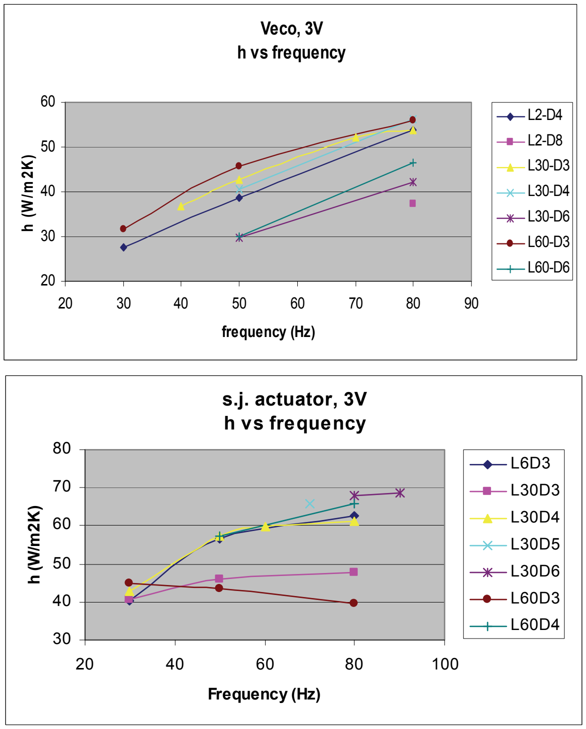

Figure 9: Comparison in terms of h vs. frequency between Veco and typical actuator

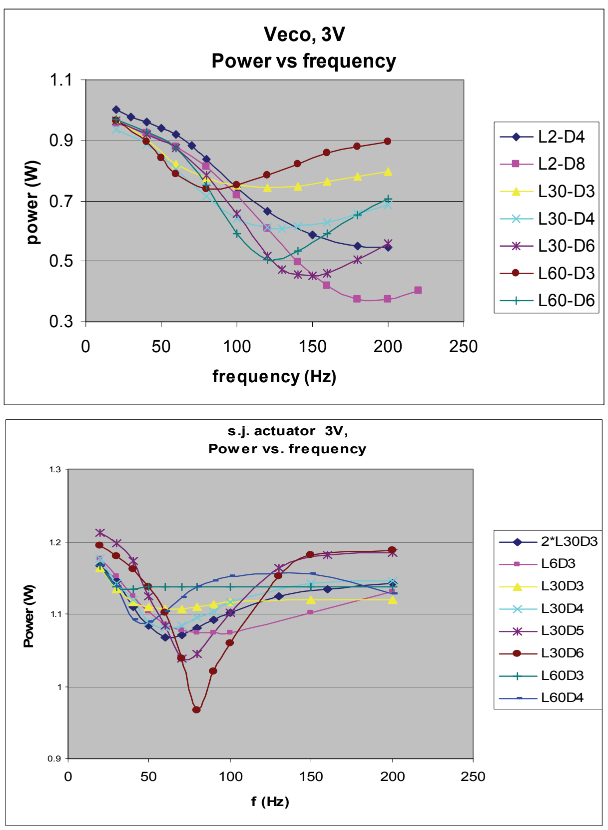

Figure 10: Comparison in terms of power vs. frequency

different because the stroke length (defined as velocity divided by frequency) becomes of the order of the pipe length. However, in this case the pulsating flow cause considerable boundary layer thinning. We measured an order of magnitude higher exit velocities when comparing synthetic jets with fans (and blowers to a lesser extent) when attaching a 2 m long pipe of 6 mm diameter.

By combing the asymmetric dipole with long pipes we found some interesting phenomena.

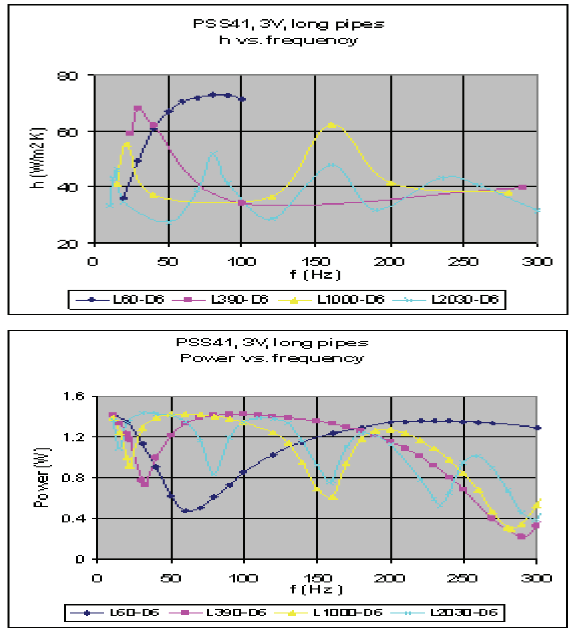

Figure 11: h and power vs. frequency for various lengths, diameter fixed at 6mm

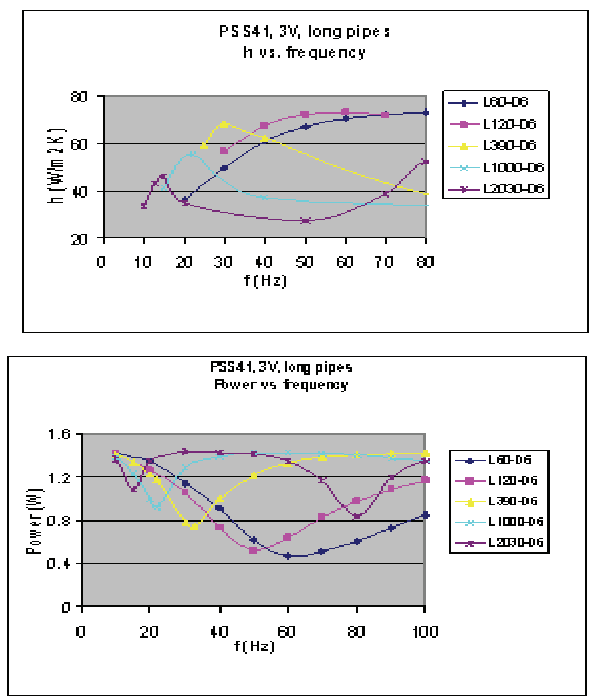

Figure 12: Limited frequency range for h and P vs. frequency

In Figure 11 h and power are shown vs. frequency for a fixed pipe diameter of 6mm and lengths varying from 60 to 2030 mm. Many maxima in h and minima in P can be seen, note however that frequencies beyond 80 Hz are not recommended due to noise issues. What is most interesting is the region from 10-80 Hz. The following graph shows this region in more detail.

It is clear from the graph that even for long pipes optimum and rather low frequencies exist that still exhibit promising high heat transfer at a promising low power level. As far as we know, attractive cooling results for such a low frequency range have never been published before. Figure 13 presents the results for the maximum h that can be reached for the first optimum as a function of length and voltage as the parameter.

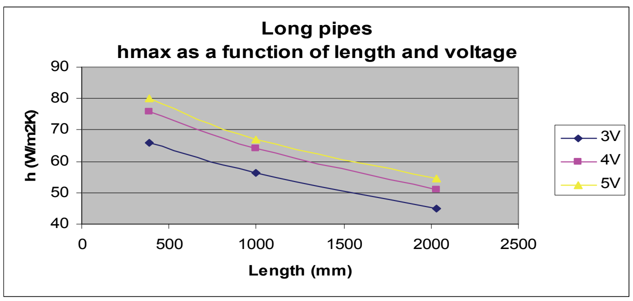

Figure 13: hmax vs. length with voltage as parameter

Observe that while the power is nearly tripled going from 3 to 5V, h is only increased by 20%, which should be at least 50%. This is an indication that increasing the voltage beyond a certain level has no effect anymore which can be attributed to two effects. The already mentioned nonlinear losses caused by separation play an increasingly important role, but also the maximum amplitude of the actuator becomes a serious constraint.

7. Lumped models for design optimisation

For the designer who wants to use synthetic jet cooling one of the main questions is how to select a proper loudspeaker. To answer this question an already existing lumped electromechanical/1D transmission line model was adapted to include nonlinear losses associated with flow separation. Starting point is the general dipole shown in Figure 2. Figure 14 gives an impression of the network. More details can be found in [3]

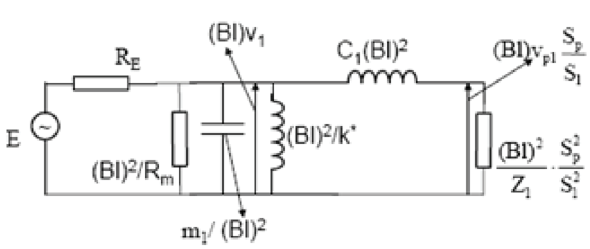

Figure 14 Acoustic model of the dipole shown in Fig.2

To demonstrate the accuracy of this model, the top graph of Figure 15 shows the power vs. frequency for a 2m long pipe for various diameters. It is striking that left of the minimum around 300 Hz the higher harmonics minima are independent of the diameter, in contrast to the right part. This is not according to 'simple' theory. After taking into account the separation losses [3], matters improved considerably, as can be concluded from the bottom graph that shows an almost one-to-one correspondence with the top graph.

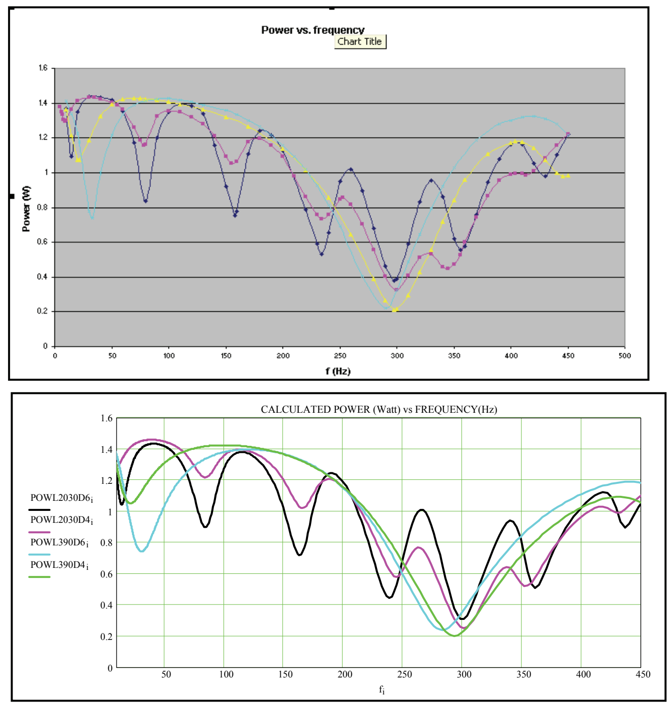

Figure 15: Power vs. frequency for a 2 m long pipe, various diameters, comparison between experiments and calculation

The model contains lumped volumes V1 and V2 (see Figure 2) coupled to a transmission line model for the pipes, where the separation losses are considered to be frequency independent. It turned out that the model performs very well for the following situations:

- relatively large volume V1 and short pipe length: 'normal' Helmholtz resonator
- Helmholtz resonator' with large pipe lengths
- symmetric dipoles
- asymmetric dipoles, with a small volume V1 compared to V2, and pipe length < 0.125*frequency

The last condition ensures that the assumption of incompressibility is fulfilled. What we want is to realise the lowest possible system resonance frequency. The model provides a quantitative analysis of how to reach this goal. If the asymmetric dipole provided with short pipe lengths is well-designed, the result is that the air is moved back and forth over the whole length of the pipe (like fireplace bellows).

8. Noise measurements

The noise as experienced by a customer is one of the most important design parameters that should be controlled. Unfortunately, it is also the most difficult one to address. While noise measurements in the available acoustic test rooms are relatively easy to perform, the challenge is to first address the root causes followed by measures to reduce the individual components of the noise. For example, in our case turbulent noises are predominant below 50 Hz and can be reduced by clever shaping of the orifice. Noise cancellation measures, both active and passive, are other options. The following table summarises the findings for the PSS41.

D	L	f	P	h	sound power
mm	*mm*	*Hz*	*W*	*W/m2K*	*dB(A)*
4	30	30	1	65	37
4	30	30	0.4	53	32
4	120	35	0.4	55	31
6	120	50	0.2	57	29
6	390	33	0.7	68	30
4	6	30	0.3	52	46
6	6	80	0.2	61	46
8	6	80	0.1	47	46
8	6	80	0.3	63	52

Table 1: Summary of single jet results, most promising embodiments for PSS41, noise data are sound power data

In red are the values that are considered 'best in class'. As one will notice there is no combination with only reds, hence it is the application that prefers either lowest power or maximum heat transfer or lowest frequency that should guide the choice. Nevertheless, some trends are worthwhile to mention. Decreasing the voltage from 3 to 2 V (compare #1 and #2) more than halves the power, decreases the heat transfer by only 20%, and the sound power goes down by 5 dB(A). Comparing #3 and #4, the difference being the change from D=4 to 6 mm, it is remarkable that the noise is lower despite the higher frequency, probably caused by the significant reduction in power. Notable is the strong increase in noise for all cases where no pipes are attached (last four cases, with L=6 mm), apparently independent on the frequency and power. The noise is primarily attributed to turbulent noise created by the very short pipe with its sharp edges. Further research should reveal how optimisation of a number of relevant parameters could decrease the noise while keeping the heat transfer performance.

We may conclude that we have achieved promising results with the PSS41 loudspeaker, especially when relatively long pipes are allowed in the application.

9. Standardisation proposal

A designer must be able to compare cooling technologies. Unfortunately, this is not trivial because a fan differs in many respects from a synthetic jet. A discussion has been started in 2007 with Prof. Glezer from Georgia Tech with the aim to complete a proposal for standardisation. A draft version was compiled in October 2007.

The basic topics of the proposal are:

- standards are proposed for noise, pressure, volume flow.
- a guideline is proposed for heat transfer, making a distinction between normal impingement and channel flow.
- rationale for the need for a Compact Thermal Model of a generic synthetic jet for the purpose of system level thermal analysis.

Clearly this effort deserves a follow-up.

10. A few remarks about the history of synthetic jets for cooling

The synthetic jet as it is nowadays applied for cooling is over 50 years old, as is clear from the following picture presented in a paper by Ingard and Labate[1]. The use of an impinging jet for cooling purposes is well-known for those skilled in the art.

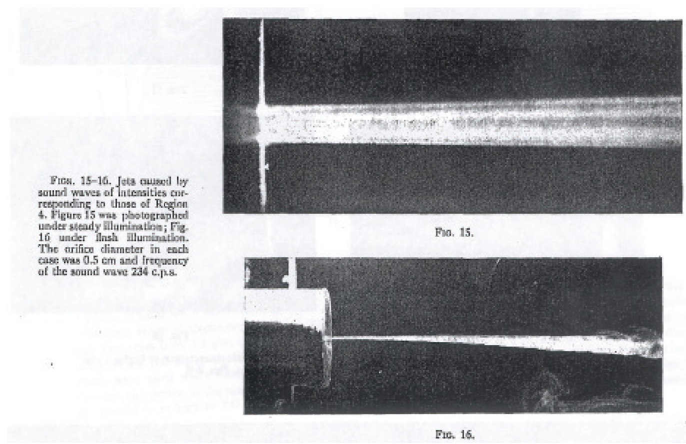

Figure 16 A clear picture of a synthetic jet created by a sound wave *(from Ingard & Labate, op.cit.)*

Dr. Arik from GE Research pointed the first author at a number of early papers published by researchers of the Technion-Israel Institute of Technology. For example, in a paper published by Gutmark et al. in 1981[4] it is stated: "In order to study this phenomenon [*heat transfer from a heated flat surface*] in a controlled flow field Gutmark et al. used a round jet that could be excited by a loudspeaker in a predetermined frequency." …..”An interesting observation was that the cooling effect of the acoustic flow was dominant…..” In another paper from 1982 by the same group[2], Figure 1 clearly shows a loudspeaker (a Philips AD12100/HP8) and a nozzle cooling a channel. Further they state: ”In the present research we explore the possibility of generation efficient acoustic cooling in two-dimensional ducts.” (Underline by the authors). A later paper published in 1986[3] is even more specific: "The paper deals with heat transfer from a train of annular vortices to a flat plate normal to it. …..Earlier work showed that periodic vortex flows may be produced by a loudspeaker, and may be used to enhance heat transfer from solid walls." Figure 1 in this paper shows the experimental set up consisting of a loudspeaker, a resonance chamber and a nozzle.

11. Further research

Despite the promising results, many questions still need to be answered due to the complex nature of the underlying physics. One of the most important questions is: can we extract design rules that are both simple and make sense, linking important parameters such as heat transfer coefficient, pipe diameter and length, frequency, driving power, air speed and noise?

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, fluid dynamics 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. 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

Earlier results acquired in 2006 show both the complexity and the potential of this relatively new cooling technology. One of the main conclusions was that for areas smaller than about 40 cm^2 the synthetic jet outperforms a standard axial fan, at least for normal impingement, both from a heat transfer and an acoustic point of view. An important design objective was to lower the frequency for which the driving power is minimal, which was the main goal for the continuation of the research in 2007. A number of loudspeakers and one actuator dedicated for synthetic jet cooling were tested over a large parameter space and compared to each other. The main conclusions can be summarised as follows.

- We have achieved promising results with an asymmetric dipole driven by a 50 mm loudspeaker, especially when relatively long pipes are allowed in the application. For long pipes in the range of 100-2000 mm it was found that rather low optimum frequencies (<35 Hz) exist that still exhibit promising high heat transfer at a promising low power level. As far as we know, these low frequencies have never been explored for cooling by other research groups. Generalisation of the conclusions with respect to other loudspeakers is still lacking.
- When maximum heat transfer is not required the coolers operating at 2V have a clear advantage over coolers operating at 3V in terms of halving the power while reducing the heat transfer by only 20%, indicating the severe nonlinearity of the actuators studied.
- When it comes to understanding the basic principles of the behaviour of the actuator devices from an *acoustical/fluid dynamics* point of view, significant progress can be reported. For example, choosing an optimal loudspeaker given certain design constraints is very well feasible. Unfortunately, from a *thermal* point of view, simple design rules are not to be expected for some time because of the many parameters that play a role.

Finally, it should be stressed that the maximum benefits of synthetic jet technology can only be acquired by completely rethinking the thermal management concept because the implementation of jets is not just about replacing fans. While of course every design change due to introduction of a new cooling technology causes headaches by designers, changing fans/blowers to synthetic jets requires a thorough understanding of the consequences. Crucial differences are: synthetic jets allow for more design freedom compared to axial fans, their much smaller exit area leads on the one hand to higher local heat transfer but on the other hand to much less mass flow, and separation of the inlet and exit flows is not trivial.

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Note: a more extended literature overview is given in reference 1.

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