Noise and Cooling in Electronics Packages

Richard H. Lyon and Arthur E. Bergles

***Abstract* - The noise produced by cooling air passing through electronics packages arises from two sources. One source is the noise of the air-moving fan of either an axial or centrifugal type. This noise may have both tonal and random components and is strongly dependent on the way that the fan is placed in the unit and on where its operation is on the fan's operating curve. Often, fans are the dominant noise sources. The flow of air produces random noise due to the turbulent generated throughout the unit. Because the turbulent airflow is also responsible for heat transfer between the components and the air stream, we can regard this part of the noise as the irreducible noise due to cooling. If fan noise were eliminated, this part of the noise would remain. There is a relation, therefore, between the irreducible noise and the cooling of the unit. But the fan noise must also be considered. The relation between total airflow related noise and cooling requirements is developed in this paper for the irreducible noise.**

***Index terms* – Cooling and noise, electronic packages, fan and flow noise, circuit board cooling**

I. Introduction

That there is a relationship between heat transfer and noise is of course well known by practitioners in acoustics and heat transfer. The primary manifestation of the relationship has been the acoustic signature a particular mode of heat transfer – pool boiling [1], well known to anyone with a teakettle.

The present discussion deals with single phase flow, and the noise is an undesirable byproduct of forcing cooling air through electronics packages. Some work has been done on parallel plate heat sinks [2], but the focus here is on ducts with wall mounted microelectronics components.

Concern for and design that includes both noise and cooling is common in industry [3]. But we believe that there has been an inherent assumption that a better design can always result in less noise. In this discussion, we indicate that there is an inherent and irreducible amount of noise due to forced air cooling that cannot be reduced by better design because turbulent flow is at the core of both heat transfer and noise generation. It is our purpose here to quantify that relationship.

Two types of flow passages through electronic packages are considered in this paper. One type is pipe or tube flow, passages that may be regarded as generally cylindrical. The sound produced by turbulent airflow through pipes has been related to volume flow and head loss. Gordon has shown such a relationship for flow in ducts based on experimental results, normalized by theoretical dependences on flow velocities and dimensions. [4,5] The average head loss is related to the fluctuating forces due to the generation of turbulence, and those forces in turn are related to acoustic dipoles that generate sound.

The second type of passage is represented by flow between parallel plates which models the flow between circuit boards. In this case, an acoustical model based on the same source mechanism as used by Gordon, dipole sound from fluctuating drag but differing because of the baffling provided by the boards, is developed here and used to predict the sound radiation between circuit boards.

Moving the air through the system requires a fan which also produces noise. Theoretical formulations for fan noise exist, but empirical relations have more credibility since they are based on the noise in typical installations. We use a published empirical relation in this paper for our estimate of fan related noise for HVAC fans that have well-controlled inlet and exhaust flow conditions. [6] Fans operating off the proper design point with disturbed upstream or downstream flow conditions will produce noise in excess of the values used here.

The same basic mechanism of turbulence generation in the passages has been shown to control the heat transfer between the walls of a pipe and the turbulent flow within the pipe. The very extensive experimental work on this problem has been summarized by McAdams in some detail [7]. A correlation for heat transfer appropriate to racks of circuit boards is presented here based on newer experimental work [8]. As we shall discuss, the work in this area has not been normalized in the same manner that McAdams followed.

Since the dipole sound due to head loss and cooling due to turbulent flow heat transfer in both types of passage are dependent on the same phenomenon (turbulent flow), there must be a relation between heat transfer and noise. One cannot have heat transfer without generating noise if the flow is turbulent, as it will be for most cooling applications. This relationship does not hold for the fan generated sound directly, but there is an indirect relationship, since fan noise is related to head gain (which equals head loss) and volume flow. In the discussion here, we will be concerned with the sound that is associated with the turbulent flow through the unit, both due to head loss in the passages, and to head rise in the fan.

II. Sound due to Turbulent Flow through Cylindrical Passages

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The derivation in this section is somewhat different from that of Gordon [4,5] in one important way. An acoustical dipole (due to localized fluctuating drag) in a confined tube produces sound power in proportion to the fluctuating drag force. The passages in a piece of electronic equipment have very porous walls – like pegboards or grillages - so we will use the "free space" relation between sound power and the *time derivative* of the fluctuating drag force.

To calculate the sound power we imagine the air passages to be generally cylindrical (but not necessarily circular) and have a hydraulic diameter $D = \sqrt{4A/\pi}$ where A is the cross sectional area of the passage as shown in Figure 1. We assume that the fluctuating drag forces have a correlation length equal to D so that each "cell" of the passage, D long and A in cross section, radiates sound as a dipole [5]:

$$\Pi_{cell} = \frac{\langle \dot{F}^2 \rangle}{12\pi \rho c^3} = \frac{\pi f^2 \langle F^2 \rangle}{3\rho c^3} \quad , \qquad (1)$$

where ρ is the density of the fluid (air) and c is the speed of sound. The characteristic (Strouhal) frequency of the sound is

$$f = \frac{V}{D} = \frac{4Q}{\pi \rho D^3} , \qquad (2)$$

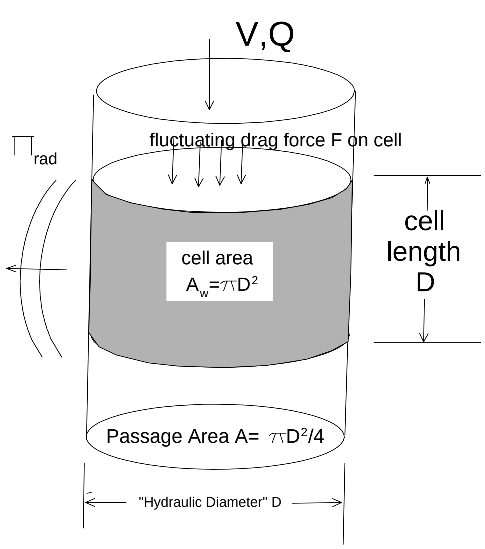


Fig. 1. Sketch of Pipe Section with Sound Radiated by Fluctuating Drag Force

where V is the average flow velocity in the passage and Q is the mass flow rate in kg/sec.

The fluctuating drag force F in a cell is related to the dynamic head of the flow through a drag coefficient C_D:

$$F = \frac{1}{2} \rho V^2 A_w C_D = \frac{8 C_D Q^2}{\pi \rho D^4} , \qquad (3)$$

where $A_w = \pi D^2$ is the area of a cell wall (note that while it is customary to relate body drag to frontal area, we are relating the friction drag to the area of the drag surface in this work). The radiated sound power from each cell is therefore

$$\Pi_{cell} = \frac{1024 Q^6 C_D^2}{3\pi^3 \rho^5 c^3 D^{10}} . \qquad (4)$$

In a passage of length L, there are L/D of these cells, each radiating independently since the fluctuating drag forces in the cells are uncorrelated. Therefore, the total sound power per unit length of the passage is

$$\Pi' = \frac{\Pi_{tot}}{L} = \frac{1024 Q^6 C_D^2}{3\pi^3 \rho^5 c^3 D^{11}} . \qquad (5)$$

This result shows the expected dependence of Q^6 that we would expect for dipole sound.

Using Eqn. (5), we can determine the relation between passage diameter D and mass flow Q for a given amount of sound radiation Π'. We use C_D=0.1 which is appropriate for fluctuating aerodynamic forces in separated flow [12]. The relation is:

$$Q = 30.8(\Pi')^{1/6} D^{11/6} \qquad (6)$$

This relation is graphed in Figure 2 for various amounts of radiated sound power.

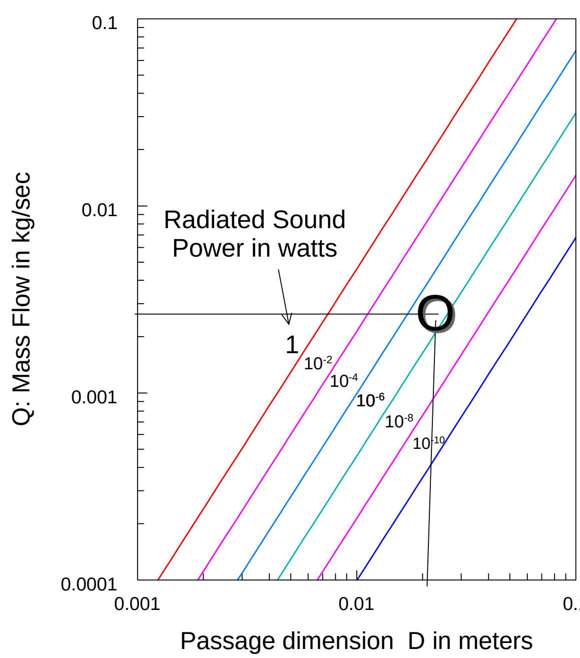


Fig. 2 Noise due to Flow through Cylindrical Passages (the point labeled O is discussed in Section VII)

III. Sound due to Turbulent Flow through Two-Dimensional Passages

In Section II, we have assumed the sound is generated in a pipe but the pipe walls are porous enough so the sound can escape through the walls. Consequently, we have used the free space formulas that relate forces to dipole radiation. In the case of flow between parallel plates that model circuit boards, the radiation is two-dimensional. As we shall see, that difference in geometry has an effect on the magnitude and spectrum of the sound.

When the obstacle producing drag force F lies between two parallel reflecting plates as shown in Figure 3, the plates reflect a line of image sources. The sound pressure at distance R from such a line at frequency f is given by

$$p(R) = \frac{\omega F \cos\theta}{4cb} H_O(kR) \tag{7}$$

where b is the distance between plates. The intensity of this sound (using the asymptotic form for $|H_O(kR)| \approx \sqrt{2/\pi kR}$) becomes

$$I(R) = \frac{<p^2>}{\rho c} \approx \frac{\omega^2 <F^2> \cos^2\theta}{8\pi\rho c^3 b^2 \kappa R}. \tag{8}$$

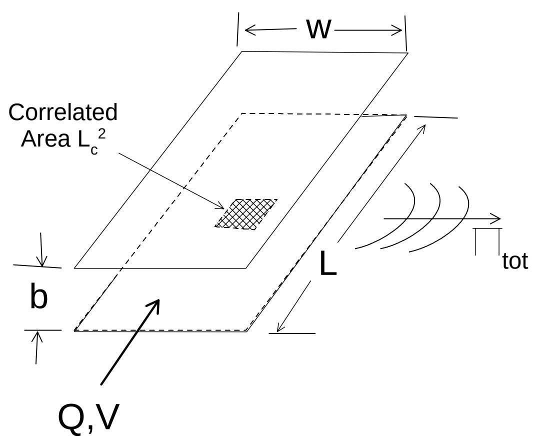


Fig. 3. Sketch of Circuit Board Passage with Sound Radiated into a Two-Dimensional Space

and the radiated power from this one coherent patch of force is then

$$\Pi_{patch} = \frac{\omega <F^2>}{8\pi\rho c^2 b^2 R}\int_0^{2\pi} \cos^2\theta\, bR\, d\theta = \frac{\omega <F^2>}{16\pi\rho c^2 b} \tag{9}$$

The number of such patches in a passage length L is wL/A_{corr}, where $A_{corr} = L_c^2$ is the correlation area of the drag force. As in Section II, the fluctuating force is related to a drag coefficient C_D: $F = \frac{1}{2}\rho V^2 A_{corr} C_D$. The total sound power per unit passage length is therefore

$$\Pi'_{tot} = \frac{\Pi_{tot}}{L} = \frac{\pi}{16}\frac{C_D^2}{\rho^4 c^2}\frac{L_c}{w^4 b^6} Q^5 \tag{10}$$
$$= 4.5x10^{-13} Q^5 / D^9 \text{ W/m.}$$

where an aspect ratio $w/b \approx 30$, and correlation length $L_c = b$ have been assumed. This result is graphed in Figure 4 for various values of radiated sound power.

IV. Cooling Fan Noise

In addition to the noise due to turbulent flow through the passages of the product, which we have related to head loss in the flow, there is noise produced by the fan that powers the flow. This component is, of course, related to the rise in head p across the fan and to the mass flow rate Q. Although there are theoretical analyses of fan noise, the complications of flow through the fan are such that we choose to use an empirical formula , based in part on theoretical expectations. A familiar formula is presented in the handbook by Beranek (converted here to SI units) [6];

$$\frac{\Pi_{tot}}{\Pi_{ref}} = 50.4\, qP^2 \text{ (air @ STP)}, \tag{11}$$

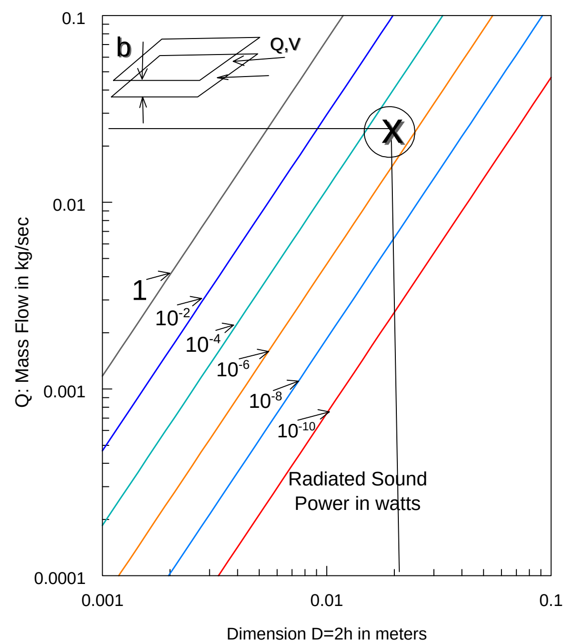


Fig. 4. Noise Radiated from Circuit Board Flow Passages (the point labeled X is discussed in Section VII)

where P is the head rise in the fan in pascals, $q = Q/\rho$ is the volume flow rate in m^3/sec, and Π_{ref}=1pW is the standard reference for sound power.

If we assume C_D=1 for the average drag coefficient

$$P = \frac{C_D \rho V^2 A_w L}{2AD} = 2.75\frac{LQ^2}{D^5},\ q = \frac{Q}{\rho} = 0.85Q. \tag{12}$$

The density is introduced here as a constant because it is already embedded in Eqn. (11). With these quantities defined, the estimate for total fan noise power is then

$$\Pi_{tot} = 323\Pi_{ref} L^2 Q^5 / D^{10}. \tag{13}$$

In Eqn. (13), sound power depends on the square of L, which is different from that in Eqn. (5). We can still compare the results if we select a passage length of $L=1$ meter. In that case, the relation between D and Q is

$$\begin{aligned} D &= 0.112\, Q^{1/2}\, \Pi'^{-1/10} \\ Q &= 79.1\, D^2\, \Pi'^{1/5} \end{aligned} \quad . \qquad (14)$$

but we remember of course that as the passage length is reduced, the fan noise decreases more rapidly than the passage noise.

Although the exponents in Eqns. (6) and (14) are not identical, they are very close. Since the coefficient in Eqn. (6) is larger than that in Eqn. (14), we might expect in most cases for the flow noise to exceed the fan noise. That will be particularly true for shorter passage lengths because of the L^2 dependence in the fan noise. Sound power graphs similar to those of Figure 2 are presented in Figure 5 for the fan noise prediction.

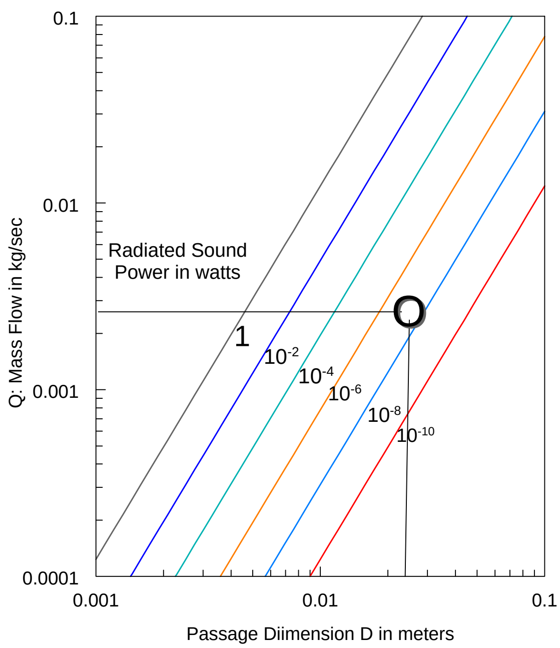


Fig. 5. Fan Noise vs. Mass Flow and Passage Dimension (the point labeled X is discussed in Section VII)

V. Cooling Due to Turbulent Flow through Cylindrical Passages.

McAdams gives the following dimensionless formula for the transfer of heat between smooth pipe walls and the flowing turbulent contained fluid: (Nusselt number=constant x Reynolds number x Prandtl number) [7]

$$\left(\frac{hD}{k}\right) = 0.023 \left(\frac{DG}{\mu}\right)^{0.8} P_r^{0.4} \qquad (15)$$

where the Prandtl number $P_r = c_p \mu / k$, μ is the fluid shear viscosity, $G=Q/A$ is the mass flow rate per unit area, k is the thermal conductivity of the fluid, and h is the heat transfer coefficient.

For air, the following quantities have values in SI units converted from English units as provided by McAdams; μ=1.85x10^{-5} kg/m-sec, k=2.6x10^{-2} J/m-sec-K, and c_p=1007 J/kg-K. These combine to give a Prandtl number of 0.707. The ambient fluid temperature is T_o and the component or "wall" temperature is T_w. The exit or exhaust temperature is T_F, leading to a temperature rise equal to

$$T_R = T_F - T_o = H_{tot} / c_p Q\,, \qquad (16)$$

where H_{tot} is the total heat flow to be carried away by the fluid.

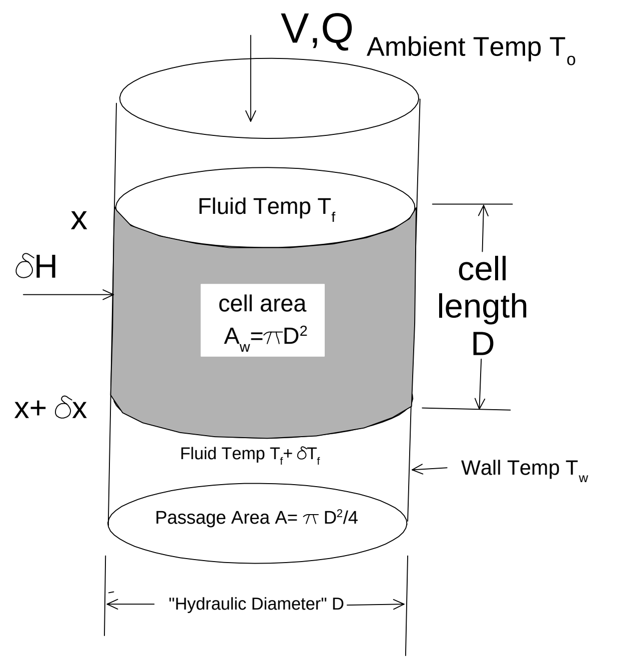


Fig. 6. Sketch of Passage Cell used in Calculating Cooling

We relate the final or exhaust temperature T_F to the flow Q and passage dimension D. by reference to the passage element shown in Figure 6. The heat flow into the element δH and the rise in the fluid temperature δT_f are related by

$$\delta H = h(T_w - T_f)\pi\, D \delta x = c_p Q \delta T_f\,. \qquad (17)$$

This expression is integrated over x from 0 to L and over temperature T_f from T_o to T_F to give

$$\begin{aligned} &T_F = T_w[1 - e^{-\beta L}(1 - T_o / T_w)], \text{or} \\ &\beta L = -\ell n(1 - T_R / \Delta T) \approx T_R / \Delta T = H_{tot} / Q c_p \Delta T \end{aligned} \qquad (18)$$

where $\beta = \pi D h / c_p Q$ and $\Delta T = T_w - T_o$. The approximation will hold for $T_R/\Delta T \le 0.1$.

In applications, the temperature rise $T_R = T_F - T_o$, the heat to be released H_{tot}, and the difference between the component and ambient temperature T_w-T_o≡ΔT are likely to be known or specified. The required mass flow rate is then also determined according to Eqn. (17). For example, if H_{tot}=50 watts, and T_R=5 K, then Q=8.87x10^{-3} kg/sec (or 16.2 cfm). With these parameters known and the dimension D defined by the layout of parts, one can estimate the noise generated by turbulence using Eqns. (5) and (13).

We can also use Eqn. (18) in the design process to develop a relation between D and Q for a given amount of heat to be dissipated per meter of passage length:

$$Q=0.04\, D^{3/2}\, (H_{tot}/\Delta T)^{5/4}. \tag{19}$$

We have chosen L=1m as a standard reference for both sound and heat transfer because of the different dependences of fan and passage noise on length L. We can use Eqn. (20) to graph D vs. Q for various values of H_{tot} ,as shown in Figure 7.

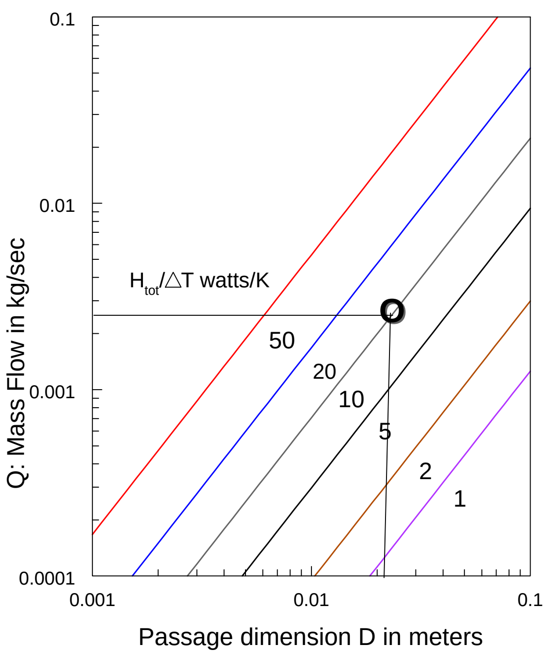


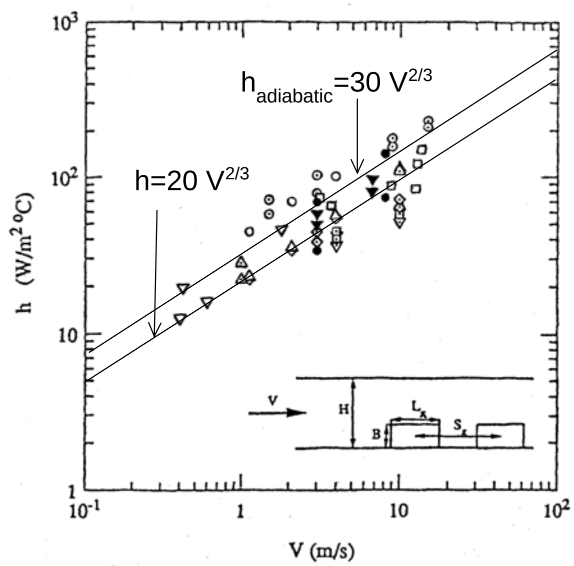
Figure 7. Required mass flow for a given passage dimension and heat flow to temperature rise ratio (the point labeled O is discussed in Section VII)

VI. Cooling Due to Turbulent Flow through Two-Dimensional Passages Simulating Circuit Boards.

A recent collection of heat transfer data by Anderson and Moffat [4] shows a dependence between heat transfer and velocity h~$V^{2/3}$. Their reported data and the correlation we are using are graphed in Figure 8. The linear regression shown is actually lower than the average of the data points. This correction is applied to compensate for the fact that the measured results correspond to a so-called "adiabatic" heat transfer coefficient.

The adiabatic heat transfer coefficient is defined in terms of the difference between an element's temperature and its own unheated temperature. Use of this parameter facilitates correlation of the data for arbitrary distributions of heating. The foregoing temperature difference is less than the conventional element-to-bulk-fluid temperature, due to thermal stratification (inadequate mixing). Thus, the adiabatic heat transfer coefficient is higher than the conventional heat transfer coefficient – by about 30% according to Moffat [9]. Accordingly, the average of the data in Figure 8 has been reduced by that amount. The resultant curve is actually quite similar to that used by IBM (interpreting h as the conventional coefficient) [10].

We can convert that correlation to a form consistent with our other relations that relate mass flow rate Q and dimension $D=2b$ to the ratio of heat flow to the temperature difference between component and also to the ambient temperatures $H_{tot}/\Delta T$ as presented in Eqn. (18). In the present case, the parameter $\beta=hw/c_pQ$ where the span of the passage w has replaced the circumference of the cylindrical passage



Name	B(cm)	H/B	S_x/B	L_x/B	
▽ Lehmann and Wirtz (1984)	1.25	2-6	1	5	2D ribs
□ Arvizu and Moffat (1981)	1.27	2-7	3-11	2	2D ribs
▽ Chang et al (1987)	2.0	1.5-3	3.5	3	single elements
⊡ Chang et al (1987)	2.0	1.5-3	6.5	3	single elements
◇ Chang et al (1987)	2.0	1.5-3	–	3	single elements
△ Wirtz and Dykshoorn (1984)	0.635	1.25-4.6	8	4	flat packs
◇ Sparrow et al (1982)	1.0	2.67	3.33	2.67	flat packs
△ Mendes and Santos (1987)	1.0	2.67	3.33	2.67	flat packs
○ Mendes and Santos (1987)	2.0	1.33	1.66	1.33	flat packs
● Anderson and Moffat (1990)	0.95	1.5-4.6	11.33	10	flat packs
▼ Anderson and Moffat (1990)	1.27	2.25-4.6	3	1	cubes
⊙ Hollworth and Fuller (1987)	0.625	2-3	8	4	staggered array

Fig. 8. Figure 1 of Reference 4 showing the data used for the heat transfer correlation in Eqn. (20)

πD. Using $Q=\rho Vwb$, and the empirical relation $h = 20V^{2/3}$ [4] leads to a relation between mass flow rate Q and dimension $D=2b$,

$$Q = 1.7x10^{-3}\, D^{1/2}\, (H_{tot}/\Delta T)^{3/2}, \tag{20}$$

where we have assumed an aspect ratio for the flow passage

$\alpha = w/b = 30$ as for the noise calculations. The result in Eqn. (16) is graphed in Figure 9. The obvious difference in form between the relationships in Figures 7 and 9 for pipe and plate flow respectively is a result of the weaker dependence on heat transfer and flow velocity in the data of Reference 8 as compared to the McAdams correlation.

VII. The Noise and Cooling Trade Off

Having related both heat transfer and sound to the passage dimension and mass flow of cooling air, we can now use Figures 2, 5, and 7 to predict the noise. For example, in Figure 7 we have selected an operating point "O" that corresponds to a passage diameter of 2.2 cm. We have also specified that the ratio of dissipated heat to temperature rise be a factor of 10, ie, for every 10 watts of heat to be dissipated, a temperature rise of 1 K is allowed. That requires an air flow of 2.6 gms/sec or a volume flow q of 10.5 cfm @ STP.

If we now enter that location "O" on the graph in Figure 2, we see that this combination of dimension and flow corresponds to a radiated sound power level of about 65 dB re 1 pW. Entering those same values into the graph in Figure 5 predicts a sound power level due to the fan of about 48 dB. In this case therefore we predict that the sound power radiated due to the flow is about 17 dB greater than the sound produced by the fan.

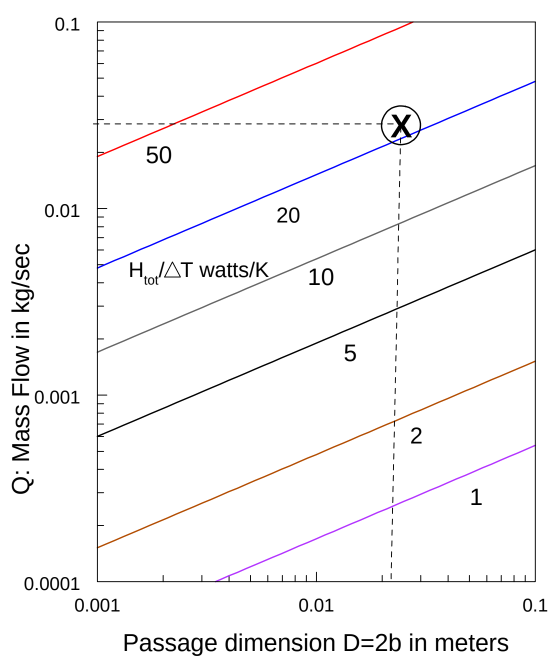


Fig. 9. Heat transfer for flow between circuit boards as a function of mass flow and separation

Suppose that for the same layout of parts (D is fixed) we want the components to run a bit cooler, so we increase the flow to 4 gm/sec (or 16 cfm). Figure 7 tells us that the parameter $H_{tot}/\Delta T$ will about double which means that ΔT is about half its previous value. How much noise penalty will we take by such a change? Figures 2, 4 and 5 all indicate about a 10dB increase in noise. Therefore the passage flow noise will continue to be dominant and the overall flow noise will increase to a power level of about 75 dB.

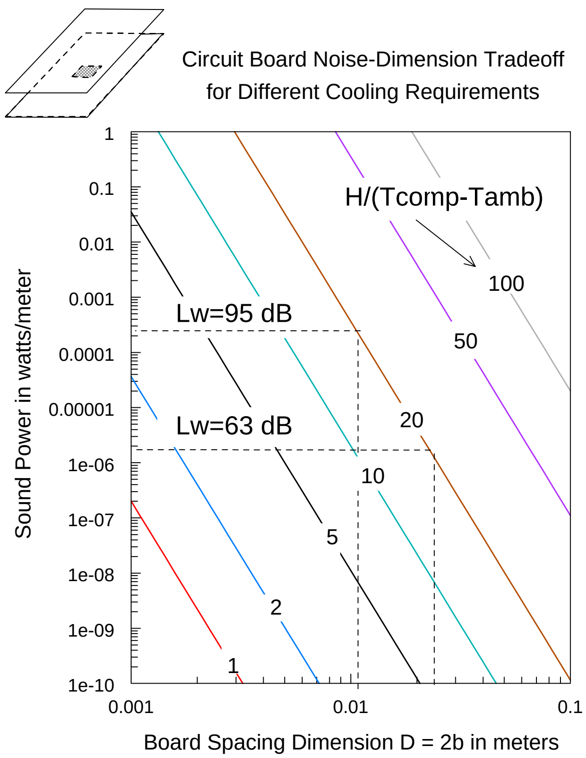
Another design issue relates to product size. We have seen for example the sharp reduction in the size of digital projectors in the past few years. The primary heat source is a lamp and temperature critical components (color wheels, LCD, DLP, mirrors, and processing chips) are in close proximity. Since most of the heat released in those products is related to the lamp, the heat to be released does not drop significantly as the size is reduced because the brightness of the light on a screen or the size of the image should not be reduced. The lamp, the image producing device, and the color wheel are all heated by the lamp, and the amount of cooling of these components must be maintained as the size is reduced. Since the airstream will be warmer, the cooling of other components is made more difficult as well. And, all this in a product that is to be quiet in a hushed conference room!

Suppose the passage dimension in the example above is reduced to 1.1 cm. Then Figure 7 indicates that ΔT is also reduced by a factor of about 2, a benefit to the cooling because of higher flow velocities in the passages. But at a price – Figures 2 and 4 indicate that the noise will increase by about 30 dB! This represents quite a challenge to the designer when the product planners request a smaller package with no reduction in performance (brightness).

The results presented here that the flow noise is stronger than the fan noise has to be taken with some caution however. The fan noise prediction in Eqn. (11) assumes that the fan is operating at or near its optimal design point and that tonal noise is not a factor. But in many applications fans operate far from their optimal design points because designers do not know the local conditions that may affect fan operation or they may not have the needed information when fan choices are made. Nearby upstream or downstream obstacles to the flow can also result in strong tones being generated that are very noticeable, even if their sound levels are not dominant in the overall sound energy.

As a final example, the sound power has been measured for a bank of 48 circuit boards with a nominal separation of $b=1.1$ *cm*. These measurements were carried out at RH Lyon Corp as part of the noise audit (analysis) of a product. The total volume flow is $Q=1.25$ kg/sec or .025 kg/sec for each channel. The component temperature is T_w=85K and the ambient temperature is T_o=25K. The total heat dissipated is 12500 W. The Reynolds number for this case is approximately 10^4. The flow path length is L=0.2 m. and the span is w=0.3 m. for an aspect ratio of $\alpha=w/b=27$. These parameters indicate a ratio $H/\Delta T$=22 and a temperature rise T_R= 9K. We see that if we enter these values for dimension and mass flow into Figure 9, we find that it predicts a value of $H/\Delta T$ close to 20.

Referring to the noise prediction in Figure 4, a sound power of about 10^{-5} w. is predicted for a 1 m. long channel. Since our channel is only 0.2 m. long, we predict a noise power of $2x10^{-6}$ w. per channel or $9.6x10^{-5}$ W for 48 channels. This corresponds to a sound power level of 80 dB. The actual sound power measured was about 89 dB. Because of the great sensitivity of the sound power prediction to the spacing value b (which is somewhat uncertain because of the board roughness), it would be possible to select a different value for *b* and achieve a better agreement. But the main point is that the agreement is reasonable and parameter dependence is retained so we can anticipate the effects of design changes on both sound radiation and component or wall temperature rise.



Figm 10. Radiated sound vs. passage dimensions for different cooling requirements

Another source of noise, not considered here, is that due to mechanical vibrations induced by the fan. These vibrations are induced by the motor and fan, and may cause rattling due to vibration of loose components or the whistling noise of flow through vent slots. But these can be reduced or eliminated by proper installation or component selection. They do not fall into the "irreducible" category that cannot be eliminated through improved design.

Finally, Figure 10 shows the result of eliminating the mass flow variable *Q* between Eqns. (10) and (20) to arrive at a relation between dimension and sound power with heat flow as a parameter for flow between circuit boards. For example , if the heat flow requirement is $H/\Delta T=20$, and $D=2.2$ cm, then the radiated noise per meter of travel is 63 dB re 1 pW. If we cut the dimension in half, the mass flow is reduced (Figure 7), but the sound still increases by 22 dB. It is clear there is still a considerable penalty in the irreducible noise as the dimensions of the product are reduced. Whether or not this effect is noticed will depend strongly on how much fan noise is present.

VIII Conclusion

There is a certain amount of noise that is inevitable when air is forced through a device to cool its components. The same phenomenon of the generation of turbulence in the passages that produces noise is also responsible for product cooling. Some products can rely upon natural convection to provide cooling, and in such products, the Reynolds number is likely to remain small enough so that the flow is laminar. Such products have essentially no flow noise but higher component temperatures may have to be tolerated. To paraphrase the credit card ad, "For everything else, there is forced air cooling".

More experimentation of the type represented by Figure 8 for heat transfer and the example presented in Section VII are needed to fill out the picture and determine the robustness of the predictions presented here. A measure of roughness for circuit boards and the components they carry correlated with measurements of drag, noise, and heat transfer will add greatly to the ability to design for minimum (but irreducible) noise.

The examples in Section VII show how sensitive the noise can be to the choice of air flow and allowable component temperature. An increase of air flow of 50% led to an increase in sound of 10 dB. But a reduction in size also has a penalty, not so much in cooling, but in a rapid increase in noise as the product is made smaller and more compact. The conclusions from this work will not surprise the designers of electronic equipments very much, but they may feel somewhat relieved that they may have overlooked some magic dust to be sprinkled on their designs that will make the noise go away. They will have to go to natural convection or liquid cooling to achieve that. On the other hand, if the noise from their prototype design is significantly greater than the values presented here, they may have a reasonable expectation that a lower noise design is achievable.

NOMENCLATURE

A	Cross sectional area of passage [m^2]
A_{corr}	Correlation area of force [m^2]
A_w	Wall area of passage [m^2]
C_D	Drag coefficient
D	Characteristic dimension of cross section [m]
F	Drag force [N]
G	Mass flow rate per unit area [kg/m^2-sec]
H	Heat flow rate [W]
H_o(.)	Cylindrical Hankel function
I	Acoustic intensity [W/m^2]
L	Length of flow passage [m]
L_c	Correlation length of force [m]
L_w	Sound power level [W]
P	Fan pressure head [Pa]
P_r	Prandtl number
Q	Mass flow rate [kg/sec]
R	Distance from sound source to observer [m]
T	Temperature [K]
V	Lineal fluid velocity [m/sec]
a	Channel aspect ratio
b	Spacing between parallel plates [m]
c	Speed of sound [m/sec]
c_p	Specific heat at constant pressure [J/kg-K]
f	Cyclic frequency [sec^{-1}]
h	Heat transfer coefficient [W/m_2-sec]
k	Thermal conductivity of fluid [W/m-K]
p	Sound pressure [Pa]
q	Volume flow rate [m^3/sec]
w	Span of channel [m]
x	Distance along passage [m]
Π	Sound power [W]
Π'	Sound power per meter of passage length [W/m]
β	Thermal parameter [m^{-1}]
κ	Acoustic wavenumber [m^{-1}]
μ	Shear viscosity of fluid [kg/m-sec]
ω	Radian frequency [sec^{-1}]
ρ	Density of fluid [kg/m^3]
θ	Angle between directions of force and observer [radians]

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