Energy label for acoustic silencers

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ABSTRACT

When a silencer is required, the flow through the silencer creates a pressure drop. The pressure drop is dependent on the design choices of both the silencer, and the available gap size. Pressure drop results, depending on the mass flow, into energy losses. While regulations become more strict with regard to energy losses, there is no suitable guideline for energy losses in silencers. Silencers cannot simply be compared by catalogue. A silencer is related to a specific noise problem that depends on source, and recipient. Therefore a method is required to compare the energy efficiency of silencers.

A method is proposed that incorporates these specific challenges. This method, relevant parameters, and design choices for the calculation will be explained. There will also be examples to explain the relevance of this method in real life situations. Finally the challenges of incorporating this method, into the system efficiency as a whole are explained.

Keywords: Sound, Attenuation, Energy usage, label, Reference silencer I-INCE Classification of Subjects Number(s): 32.4; 34.1

(See . http://www.inceusa.org/links/Subj%20Class%20-%20Formatted.pdf .)

1 INTRODUCTION

Energy efficiency is a high priority issue. Measures to improve energy efficiency are adopted everywhere. Energy labels guide the way to a more energy efficient solution. There are no energy labels for sound attenuators developed, even though sound attenuators effect the energy efficiency of an installation. Developing energy labels for sound attenuators, provides the opportunity to apply more energy efficient sound attenuators. By providing information on a number of annual kWh consumption, or even annual financial consumption, based on an average efficiency and average energy prices helps in making an informed decision. However, energy consumption numbers alone don't show potential energy savings, and therefore cannot prevent excessive energy consumption due to a too small available cross section area in the design criteria, or tell how the most efficient option relates to the best energy efficient solution. This article proposes a method in order to determine an energy efficient reference sound attenuator, and generate an energy label by comparing for a given sound attenuator with this reference attenuator.

In part two, the working method in determining an energy label is explained. In part three some examples are given. Part four consists of reflections, and contains additional information on where to place system boundaries. Part five contains the conclusions, and recommendations.

2 DETERMINING AN ENERGY LABEL

2.1 The reference attenuator

The energy label will be determined when the attenuators are compared to a reference attenuator, or set of reference attenuators. Sound requirements vary highly per situation. Partly because noise is always a local issue. Partly because the sound source, and positioning of the sound source could also result in a widespread variation of the required attenuation spectrum. The reference attenuators are therefore different for each noise problem, but should have the same pressure drop, no matter who calculates them.

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The reference attenuator also needs to have a relatively low pressure loss, compared to the attenuation for a wide spectrum. An attenuator with a splice distance of 100 mm, and an attenuator thickness of 100 mm, has good attenuation values compared to the relatively low pressure drop. Ideally, this attenuator could even be equipped with bulbs, and runoff profiles, see figure 1.

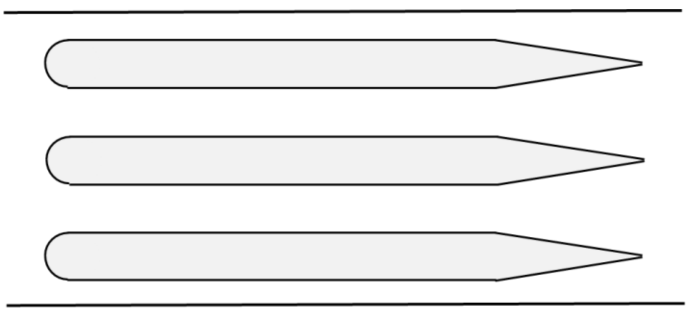

Figure 1: Attenuator with bulbs and runoff profiles

The length then is determined by the acoustic requirements. For this a table of attenuation values are provided. The attenuation is a multiple length of 250 mm, with a minimum of 500 mm. The maximum length is 4500 mm, otherwise vibrations could become the dominant noise-path.

Table 1 – Attenuation values for different lengths of the reference attenuator

mm	31 Hz	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz	16 kHz
500	1.3	1.4	1.8	3.7	9.6	17.1	19.6	18.1	12.2	9.8
750	1.9	2.1	2.7	5.5	13.9	23.9	26.8	23.0	14.3	10.0
1000	2.6	2.7	3.6	7.3	18.0	29.5	33.0	27.6	16.5	11.9
1250	3.2	3.4	4.4	9.0	21.8	33.9	38.0	31.8	18.6	12.8
1500	3.8	4.1	5.3	10.7	25.2	37.3	42.0	35.7	20.6	13.9
1750	4.4	4.7	6.2	12.3	28.3	39.8	45.1	39.1	22.6	14.9
2000	5.0	5.4	7.0	13.9	30.9	41.6	47.4	42.1	24.5	15.9
2250	5.7	6.1	7.9	15.5	33.3	42.9	49.1	44.7	26.4	16.9
2500	6.2	6.7	8.7	17.0	35.3	43.8	50.4	47.0	28.2	17.8
2750	6.8	7.3	9.5	18.4	37.0	44.4	51.3	48.9	29.9	18.8
3000	7.4	8.0	10.3	19.8	38.4	44.9	52.0	50.6	31.5	19.8
3250	8.0	8.6	11.1	21.1	39.6	45.2	52.5	51.9	33.0	20.7
3500	8.5	9.2	11.9	22.3	40.6	45.4	52.8	53.1	34.5	21.6
3750	9.0	9.8	12.6	23.5	41.4	45.5	53.1	54.0	35.9	22.5
4000	9.6	10.4	13.4	24.6	42.1	45.6	63.3	54.8	37.2	23.5
4250	10.1	11.0	14.1	25.7	42.6	45.7	53.4	55.5	38.4	24.3
4500	10.6	11.5	14.8	26.7	43.1	45.8	53.3	56.0	39.5	25.2

The reference attenuator is equipped with bulbs, and runoff profiles, as shown in figure 1. Length has a relatively low influence for the pressure drop, but to stay realistic, the maximum length is set at 4.5 m. When more attenuation is required, a second attenuator is required, with a bit more pressure drop because of a second bulb, and second runoff profile. Flow resistance ζ values can be found in table 2.

Table 2 – Flow resistance constants

part	configuration	ζ	Unit
INWARD	Duct to bulb	0.023	[-]
	Plenum to bulb	0.10	[-]
ALONG	Alongside	0.15	[/m]
OUTWARD	Runoff to duct	0.10	[-]
	Runoff to plenum	0.23	[-]

$$\partial p = \frac{1}{2}\rho \cdot \left(\sum \zeta\right) \cdot v^2 \tag{1}$$

Where ρ is the density of the medium, and v is the flow velocity in the splices.

2.2 The reference cross sectional area

Another property that influences the pressure drop is the cross section area of the attenuator. Attenuators with a large cross section have lower flow velocities, which results in significantly less pressure drop. A reference attenuator needs to have a predetermined cross section area. In reality the flow velocity in the splice of an attenuator is not always the same. It is common to find low flow velocities in (small) systems with a small flowrate, and higher velocities in a system with a higher flowrate. Therefore, the cross section area could best be related to the flowrate. Here again a working method for determination of the system boundaries, and the comprehensive flowrate φ in kg/h is required.

$$v = \frac{11.5 \cdot \log \cdot (\varphi)^2}{25} + 0.5 \tag{2}$$

Table 3 – examples of flow velocities

Flow φ [kg/h]	Velocity v [m/s]	Cross sectional area [m^2]
100000	12	3.86
10000	7.86	5.89 10^{-1}
1000	4.64	9.98 10^{-2}
100	2.34	1.98 10^{-2}
10	0.96	4.82 10^{-3}

In order to ensure low pressure losses at the reference attenuator, the splice velocities v is maximized at 12 m/s.

1.1.1 Multiple reference attenuators

To understand how a reference attenuator should look like, doesn’t mean the sound requirements can always be met with one attenuator. In some specific situations flow generated noise could be dominant. In both cases the reference attenuator consists of at least two attenuators. In the case that flow generated noise is dominant, the second attenuator needs to have a lower flow velocity than the first attenuator. Here again a method for determination of the second attenuator is required. Choose the flow velocity so, that flow generated noise is 5 dB below the noise limitation(s). Note that the flow generated noise in the first attenuator also needs to be added up to the noise coming through the first attenuator. Also note that when the reference attenuator consists of multiple attenuators, it doesn’t mean the proposed attenuator in a real life situation consists of multiple attenuators, and vice versa.

When the reference attenuator consists of more than one attenuator, the first attenuator is at least 4500 mm long. The attenuation values of the second (and third) attenuator are the same as the first attenuator, but the size of the cross section is different. When a third attenuator is necessary the second attenuator is also at least 4500 mm long, before a third is added. The velocity within the third attenuator is the average flow velocity of the first and the second attenuator, and the third attenuator is placed between the first and second. The noise then first encounters the first, then the third, and then the second attenuator. The pressure drop in each attenuator is the same as if the attenuator had the same cross section as the first attenuator. Otherwise multiple attenuators would have less pressure drop when the second attenuator is increased when flow generated noise is an issue.

2.3 Flow generated noise

The sound attenuation of a chosen attenuator in a specific system should include the flow generated noise, because a system that produces a relevant amount of flow generated noise, needs to attenuate much more sound to produce the same net amount of attenuation.

The numerical value of the flow induced sound is the same as mentioned in the VDI 2081[1)]. On one hand the sound attenuation in longer attenuator will reduce the outgoing sound power, on the other hand small unforeseen edges, connectors, and overlap could increase this value as well. The sound pressure level is [1]:

$$L_p = 7 + 50 \cdot \log(v) + Z \tag{3}$$

$$L_w = 7 + 50 \cdot \log(v) + 10 \cdot \log(S) + Z \tag{4}$$

Z values vary for each octave band, and can be calculated using the mid-frequency f_m:

$$Z = -0.55 \cdot \left(\log\left(\frac{f_m}{v}\right) \cdot 3.33 - 1.65 \right)^2 + 0.8 \cdot \left(\log\left(\frac{f_m}{v}\right) \cdot 3.33 - 1.65 \right) - 6 \tag{5}$$

The frequency weighting depends on the flow velocity. When also the A weighting is accounted for, this will result in a graph like figure 2:

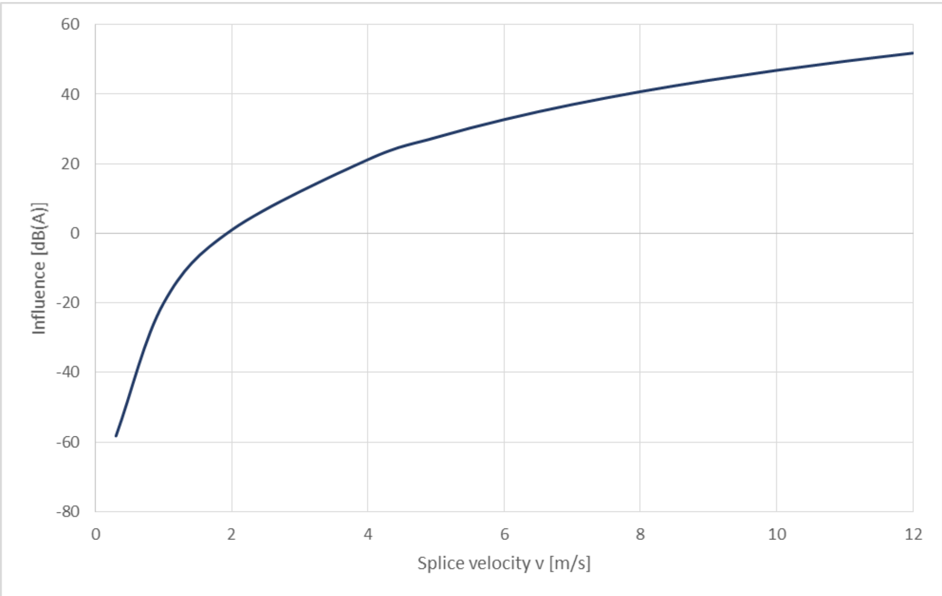

Figure 2: Influence of velocity in sound power/presure in dB(A)

Figure 1 is given in order to make a quick estimation on whether or not regenerated sound is relevant. When an attenuator is part of a more complex system, the spectral contribution of flow induced sound needs to be accounted used in the calculations.

In order to determine the sound power, add 10*log(S) to the spectral sound pressure level. Where S is the cross section surface area.

2.4 Energy label assessment

The reference splitter attenuator has energy label A. Label C has double the energy consumption as A. 4 times as much energy consumption is E. Eight times G, and 16 times is I: half as much as A++. In table 5 a schematic is shown of the upper and lower limits of an energy label as a function of the pressure drop of the reference attenuator A. In table 5, A represents the pressure drop in the reference attenuator.

Table 5 – examples of flow velocities

Label	minimum pressure drop	maximum pressure drop
A++++	$A*2^{-9/4}$	$A*2^{-7/4}$
A+++	$A*2^{-7/4}$	$A*2^{-5/4}$
A++	$A*2^{-5/4}$	$A*2^{-3/4}$
A+	$A*2^{-3/4}$	$A*2^{-1/4}$
A	$A*2^{-1/4}$	$A*2^{1/4}$
B	$A*2^{1/4}$	$A*2^{3/4}$
C	$A*2^{3/4}$	$A*2^{5/4}$
D	$A*2^{5/4}$	$A*2^{7/4}$
E	$A*2^{7/4}$	$A*2^{9/4}$
F	$A*2^{9/4}$	$A*2^{11/4}$
G	$A*2^{11/4}$	$A*2^{13/4}$
H	$A*2^{13/4}$	$A*2^{15/4}$
I	$A*2^{15/4}$	$A*2^{17/4}$
J	$A*2^{17/4}$	$A*2^{19/4}$
K	$A*2^{19/4}$	$A*2^{21/4}$
L	$A*2^{21/4}$	$A*2^{23/4}$
M	$A*2^{23/4}$	$A*2^{25/4}$

2.5 Temperature, fluent medium, and velocity influences

When the fluent that carries the noise is other than air, or at temperatures higher than 30°C or lower than 10°C, the attenuation values D_0 are not the same as in table 1. The easiest fix is to adjust the values for the change in sound velocity. Higher sound velocities effectively shorten the effective length of the attenuator, and deliver a proportional decrease in attenuation value. The nominal sound velocity is 340 m/s. The attenuation value D per octave-band becomes:

$$D = D_0 \cdot \frac{c_0}{c} \tag{5}$$

A fluent that varies too much from air, such as liquid water, is not suitable to obtain an energy label for attenuators, through this proposed method.

Because the flow velocity in the reference attenuator is maximized at 12 m/s the influence of flow velocity, and flow direction can be neglected.

2.6 Size of the attenuator

When a requirement for noise is set at a point in close proximity to the attenuator, and is expressed as a sound pressure level, the geometry becomes important. The reference attenuator is fabricated as close to a perfect square as possible, while the width changes in steps of 0.2 m.

3 EXAMPLES

3.1 A silencer to vent a living room

In order to reduce incoming traffic noise through the ventilation opening in a living room by 12 dB an attenuator is proposed. The living room has only one external wall, facing the traffic. The air ventilation is supposed to be only 40 kg/h. A system works with an attenuator on the traffic side with flow velocities of 3 m/s inside the attenuator. The air is then extracted from the room through a mechanical ventilation system that is connected to a side of the building with no relevant noise problem. The ζ value of the attenuator is 1.0, and the velocity in the mechanical ventilation is 6 m/s with a ζ value of 1.4.

A reference attenuator that solves both incoming and outgoing flow will only be able to ventilate on the side where there is traffic noise. When only the incoming attenuator is evaluated, this attenuator may be compared to a reference attenuator that provides only a solution for the air intake. The flow velocity in the attenuator is 1.68 m/s. The opening area in the reference attenuator is nearly twice as large for the incoming flow only, and needs to attenuate 3 dB more. Because the outgoing flow, and the sound through the wall is otherwise negligible. The ventilation area further increases, so the attenuation value needs to be 6 dB more. In this example flow generated noise may be neglected.

Theoretically a 1.5 m long attenuator is required with a plenum to plenum ζ value of 0.555. The energy consumption of the proposed solution is 6.6 times as high as the reference attenuator, therefore the energy label is F for the attenuator that provides the incoming flow of air.

This may seem like a low energy label, but there is another special case where the air supply occurs though natural ventilation. Let the maximum air velocities be as low as 0.5 m/s, the ζ value is 1.5, in combination with an attenuator ζ=1.8. The efficiency is supposed to be 0.8, and the price 0.23 €/kWh.

Table 6 – overview example living room ventilation

description	Velocity [m/s]	Pressure drop [Pa]	Pressure drop reference attenuator [Pa]	Energy label [-]	Annual mechanical energy consumption [kWh/year]	Annual costs [€/year]
Incoming airflow attenuator	3	3.6	0.94	E	0.29	0.08
Air ventilation system	3/6	33.8	2.14	I	2.8	0.79
Natural ventilation with attenuator	0.3	0.27	0.94	A++	0	0
Natural ventilation	0.3	0.23	0.94	A++	0	0

Not all energy labels are relevant, and the determination of an energy label may not always be required. Nevertheless this example shows the label is a logical outcome.

3.2 An attenuator for the cooling of a pump in an enclosure

In an enclosure a 500 kW pump is placed. 90% of the heat is transferred to the fluid, and 10% of the heat needs to be removed by ventilation. The maximum temperature increase is 10 ^{o}C. A ventilation of roughly 0.4 kg air per second, or 144 kg/h, is required. ρ is 1.2 kg/m^3. The sound pressure level in dB(A) acting on the wall of the enclosure is:

63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz
85	91	94	100	101	107	112	99

The sound pressure through the silencer may not exceed 65 dB(A) at 1 m. The flow velocity now is

2.6 m/s, and the area 10 by 2 cm. Due to other sound sources, and directivity, the geometric attenuation, D_{geo} is set to be only 4.1. The reference attenuator now is 1750 mm long, has a ζ value of 0.592, and the pressure drop is 2,5 Pa. A much smaller alternative has a pressure drop of 11 Pa, and therefore has energy label F.

Here again, as in example 1, flow regenerated noise is negligible. Judging by the numbers an energy label may not be required, but could become interesting if 20 times more cooling is required.

Velocity [m/s]	Pressure drop [Pa]	Pressure drop reference attenuator [Pa]	Energy label [-]	Annual mechanical energy consumption [kWh/year]	Annual costs [€/year]
3	11	2.5	F	3.2	0.92

3.3 An attenuator in an enclosure with two 3 MW diesel engines

In order this example, the sound power that acts on the reference cross section. The acoustic requirement is a sound power of 90 dB(A).

	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz
L_{w_in} / m^2	82	95	101	105	107	105	105	106
D_{filter}	0	0	-1	-1	-2	-4	-3	-2

The regenerated noise is now high, but still negligible. The outlet differs from the intake. First of all the volume flow is decreased by 10%, because this flow was used to feed the engine. Secondly the flow needs to be forced through the attenuators, so the area where the sound enters the outlet could in reality be smaller, so less sound power, while on the other hand the sound power is increased by the sound power of the fans. The reference attenuator needs to be reproducible, and therefore has no fan, or smaller opening.

Industrial prices are 0.115 €/kWh

Table 7 – overview example emergency power

description	Velocity [m/s]	Pressure drop [Pa]	Energy label [-]	Annual mechanical energy consumption [kWh/year]	Annual costs [€/year]
Reference attenuator intake	12.0	58	A	11761	1691
Reference attenuator outtake	11.8	56	A	10220	1469
Compact cheap alternative intake attenuator	14.3	215	E	43597	6267
Compact alternative outlet attenuator	13.0	175	D	31938	4591
Second alternative intake attenuator	8.8	61	A	12369	1778
Second alternative outlet attenuator	13.9	54	A	9855	1416
Acoustic design intake with bend	10.5	119	C	24131	3469

So far the examples have only come up with attenuators, and not for the acoustic design. The intake now is equipped with an additional acoustically lined bend that increases the pressure drop with 8 Pa, and has the following acoustical properties:

	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz
D_{bend}	-0.5	-1.5	-2	-3	-4	-5	-7	-6

The bend will not be included in the calculation of the reference attenuator, but will be taken into consideration in the design of the other attenuator. Now the attenuators cannot be compared, but the acoustic design can still get an energy label. When only the attenuator is replaced by a reference attenuator, the attenuator can still obtain an energy label. But to translate the requirements, mistakes are easy to occur. It is advisable to only evaluate the acoustic design. This is done in the last line of table 7, and results in energy label C for the acoustic design.

3.4 An attenuator in a concrete hall with a 2 MW diesel engine

In order to simplify this example , the sound power per area that acts on the attenuator cross section area is given.

Now let the acoustic requirement be a L_w of 55 dB(A) instead of 90. Because of other equipment the air intake is still 100000 kg/h. Now flow generated noise will be higher than the demand, so the reference attenuator needs to consists of multiple attenuators. In this case only the reference attenuators of the air intake will be calculated. The cross section is still 3.86 m^2 and the splice velocity 12m/s.

	63 Hz	125 Hz	250 Hz	500 Hz	1 kHz	2 kHz	4 kHz	8 kHz	tot	Δp [Pa]
L_{w_in} / m^2	60	72	83	91	100	101	99	97	105.5	
D_{filter}	0	0	-1	-1	-2	-4	-3	-2		
$L_{w\ in1}$	65.8	77.8	87.8	95.8	103.8	102.8	101.8	100.8	108.5	
1st Attenutor	11.5	14.8	26.7	43.1	45.8	53.3	56.0	39.5		75.6
Attenuated L_w	54.3	63	61.1	52.7	47.3	43.8	47.1	51.6	65.8	
L_w flow 1	34.9	44.5	50.3	52.9	52.2	48.4	42.2	32.9	55.9	
$L_{w\ in2}$	54.3	63.1	61.4	55.8	53.4	49.7	48.3	51.7	66.3	
2nd Attenuator	8.6	11.1	21.1	39.6	45.2	52.5	51.9	33.0		64.0
Attenuated L_w	45.7	52.0	40.3	16.2	8.2	-2.8	-3.6	18.7	53.1	
L_w flow 2	29.0	38.0	43.3	45.4	44.2	39.9	33.1	23.2	48,3	
Result	45.8	52.2	45.1	45.4	44.2	39.9	33.1	24.5	54.4	139.6

The total pressure drop of the reference attenuator than is 139.6 Pa, which has an interesting amount of annual costs given the flow of 100000 kg/h. Note that some gas turbines need an even bigger air flow.

description	Velocity [m/s]	Pressure drop [Pa]	Energy label [-]	Annual mechanical energy consumption [kWh/year]	Annual costs [€/year]
Reference attenuator intake	12.0	139.6	A	28308	4069

The entire working method described in figure 3 could be implemented in a simple tool.

4 Reflection

4.1 How does the silencer relate to the reference silencer

In all situations a reference silencer can be determined. The reference silencer is in all cases better, than the more practical silencer. Better silencers than the reference silencer are achievable when there is more available space, but could cost more material and handling.

4.2 Importance of the values of the reference attenuator

Attenuation is not always straightforward. Through vibrations, and directional effects attenuators could perform different in real life than on charts. The noise through vibrations is tackled roughly by maximizing the attenuation values. This also leads to a higher, more realistic pressure drop.

Directivity effects could be neglected in a reference attenuator. It is the responsibility of the supplier that his attenuator is designed to meet the written specifications. This means that an attenuator will need to address these effects. Addressing this effect in the reference attenuator may lead to unnecessary complications.

4.3 Relevance of the energy label, and system boundaries

To show the relevance of the energy label of the attenuator it is good practice to accompany the pressure drop through the attenuator with the pressure drop through the entire system. This primarily shows the relevance, but could also help to find the other relevant components regarding the pressure drop, and possibly improve the design.

Again, this article is to be seen as a proposition only. Nevertheless the preferred option would be to review only the pressure drop in the attenuator, and extend this label with two values, that show the relevance of the attenuator. In order to get a clean design of the reference attenuator, it was chosen not to include the acoustic effects of bends, narrowness of the channel, but to include the direct, and indirect sound on the reference cross section area, as if this was part of the room where the sound source is situated.

When filters, or rain louvres are a necessity, the acoustic properties of these parts should be included, but not their pressure drop. Theoretically they can be sized up, so they cause negligible pressure drop, but retain their acoustic properties. When filters or rain louvres are required, it is good practice to mention their pressure losses separately, because they are not part of the ‘acoustic’ design, but are part of the system design.

All other channel parts are not included in the reference attenuator, as mentioned above, because they serve no necessary function.

4.4 Instructions

An energy label can be determined in a number of steps, as mentioned in figure 4.

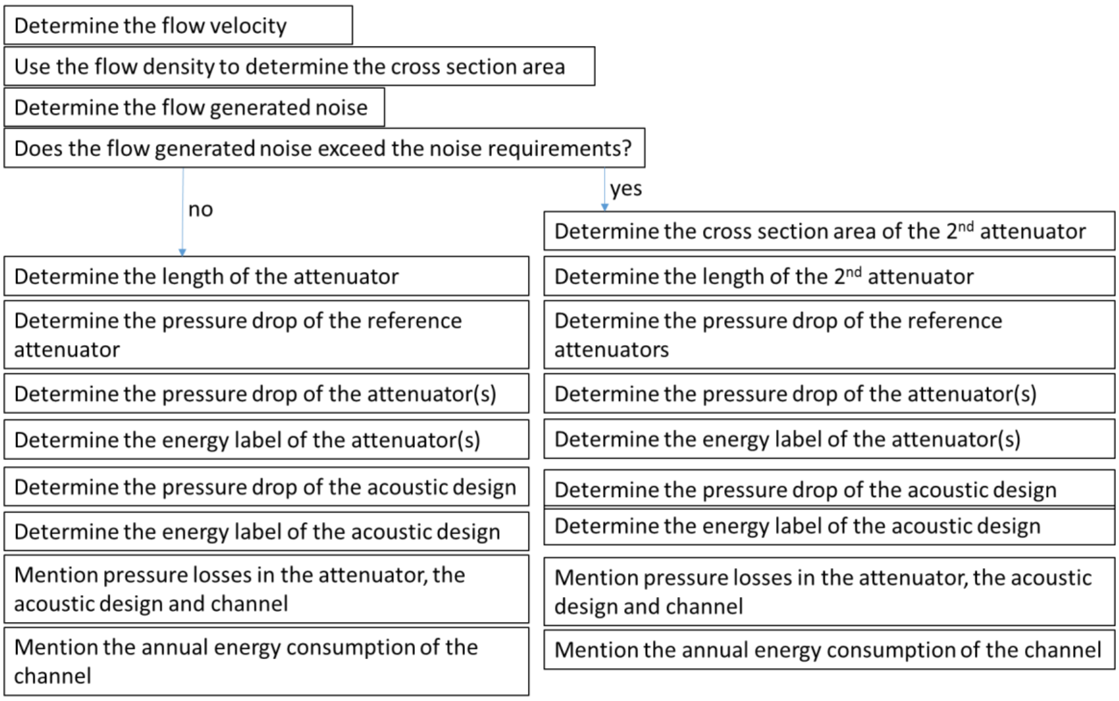

Figure 4 The steps to determine energy labels

5 CONCLUSION

It is possible to define an energy label for silencers that tells how a given silencer relates to a hypothetical energy efficient design. Together with a value to predict the energy usage of the attenuator, and the energy losses in the ventilation system, this will provide a good starting point in order to evaluate the design, and improve design choices. Such an analysis can be conducted for attenuated flow systems, using the roadmap presented in this article. However, this is a first proposal that may be subject to improvements.

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