Experimental Investigation on Acoustic Effects of Geometrical Modifications of Splitter Attenuators for Power Generation Systems

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Abstract
High end splitter-type silencers represent the object of the present paper, that summarizes the results obtained in an experimental campaign conducted in the framework of FlowAirS Project - ITN FP7, Marie Curie Actions. Sound measurements on several fully functional splitters scaled 1:3 with Mach and Reynolds number similarity to the typical conditions in gas turbine exhaust stack have been performed. Different trailing edge shapes including tapered and serrated extensions have been tested. Only moderate changes in the measured noise have been found, despite a reduction of the total pressure loss that can be achieved by appropriate design of the splitter trailing edges. Considerable effects due to the covering perforated plate have been found. The results are analyzed applying Nelson and Morfey's theory for obstacles in ducts, the scaling provides a good data collapsing, in particular for the frequencies above the cut-on.

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1. Introduction

Splitter attenuators are used in a wide range of applications, from HVAC systems to high-end silencers for industry and power generation plants. The present paper refers to this second type and summarizes the results obtained in an experimental campaign conducted in the framework of FlowAirS Project - ITN FP7, Marie Curie Actions. Usually the silencer represents the last stage for noise reduction in the exhaust stacks of power systems, so that any sound generated downstream of the silencer is radiated into the open without any relevant attenuation. As regulations demand increasingly strict noise limits, silencers in exhaust stacks need to be designed to achieve the necessary high acoustic insertion losses. With the upstream noise reduced to ever lower levels, the effect of the self-noise generated by the splitter itself - as a bluff body immersed in the flow - becomes critical for the overall performance of the silencing system. Because the typical operative conditions in exhaust stacks (e.g. high temperatures, difficult access, undefined parameters, etc.) make an in-depth study of the problem in real installations essentially impossible, a study on the acoustic optimization of the splitter geometry has been conducted under laboratory conditions.

2. Approach

The described problem has been treated from an experimental point of view. Sound measurements on splitters, equipped with several trailing edge extensions, have been performed in the Müller-BBM silencer test stand, in Planegg (Munich, Germany).

2.1. Geometry and properties of the splitter

Fully functional model splitters, configured as they typically are for operations in flue gas stacks have been constructed and manufactured by BBM Akustik Technologie, and have been used as test objects (Fig. 1, 3).

The dimensioning of these model splitters derives from the similarity of Mach and Reynolds

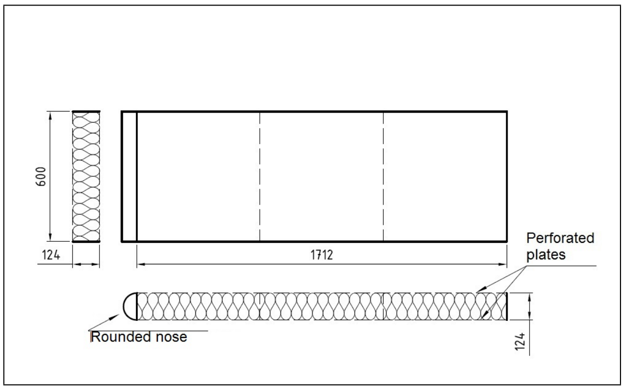

Figure 1. Sketch of the model splitter.

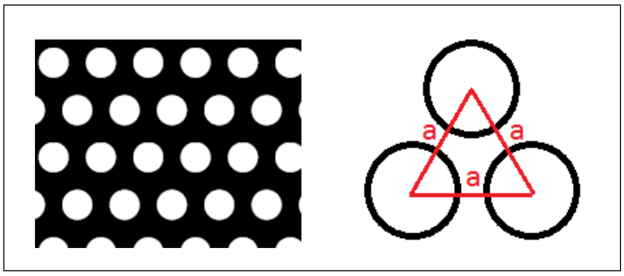

Figure 2. Perforation pattern.

numbers to the working conditions of a typical stack silencer (Mach $\approx$ 0.08, Reynolds referred to the splitter thickness $\approx$ 200000). The small scale model splitters are totally functional and similar to the real size ones with respect to absorbing material and geometrical properties, except for the covering perforated plates, that have the same perforation diameter than in the real full size splitters. The splitters are filled with sound absorbing cushions covered by the perforated plates. A steel plate can be inserted between the cushion and the perforated plate to perform tests without any significant sound absorption by the splitters. The splitter frame is equipped with a rounded nose of semi-circular shape, made by 1 mm plates of galvanized steel (Fig. 1, 3). The side walls are perforated plates with the following characteristics: 1.5 mm of thickness, a perforation diameter $d_H = 5$ mm, a distance between the holes $a = 8$ mm (Fig. 2), which corresponds to an opening ratio equal to 35.4%. The absorbing part is made by cushions in glass fiber with a density of 125 kg/m^3.

In order to allow a simple trailing edge modification, several extensions have been manufactured, that can be attached to the downstream end of the splitter. Different wooden extensions for blunt and tapered trailing edges (Fig. 4) and different metal serrated extensions have been tested.

Figure 3. Manufactured splitter.

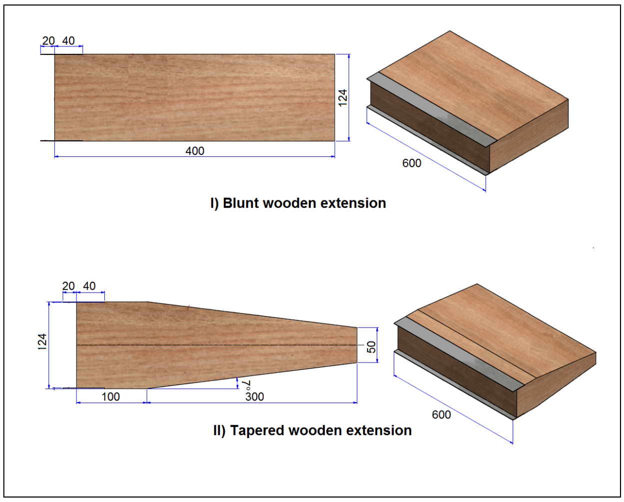

Figure 4. Wooden extensions.

2.2. Test Stand

The test rig used for the measurement campaign has been built in compliance with the guidelines of the ISO 7235:2003(E), a sketch is provided in Fig. 5. Essentially, the test stand comprises three parts: a source room (equipped with a loud-speaker and a silenced fan as a flow generator), a test section and a reverberation room. The three parts are connected through a duct with a diameter of 400 mm. Different from the sketch in Fig. 5, the connecting duct between the test section and the reverberation room includes a 90° bend and a 45° bend in the final part connecting to the reverberation room itself. The total length of the duct between test section and reverberation room is about 20 m.

In tables I and II the main characteristics of the reverberation room and of the test section are reported. Under the given conditions, in compliance with the ISO 3741:2010(E), the lowest one-third octave band

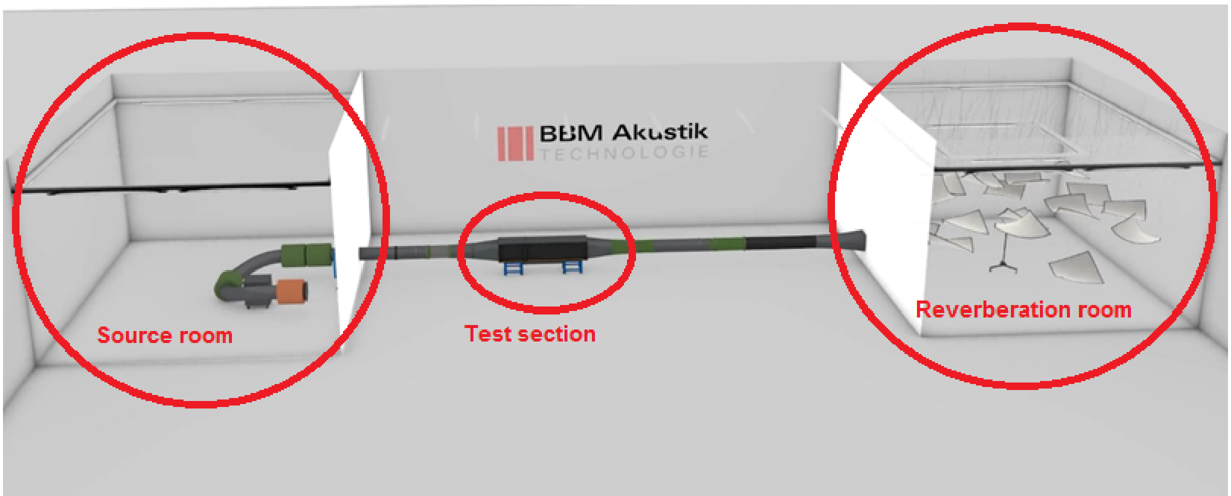

Figure 5. Silencer Test-Stand

Table I. Reverberation room characteristics.

length	[m]	8
depth/length	[-]	0.83
height/length	[-]	0.47
total volume	[m^3]	199.6
diffuser total surface	[m^2]	52

Table II. Test section characteristics.

length	[m]	2.9
width	[m]	0.275
height	[m]	0.6
cut-on frequency	[Hz]	283.3

frequency that the reverberation room is suitable for is 100 Hz.

The limiting cut-on frequency is given by the circular connecting duct at around 500 Hz, almost the double of the cut-on frequency of the rectangular test section.

A Norsonic Nor1220 microphone, equipped with Norsonic preamplifier type 1201, mounted on a rotating gallows, and a Brüel & Kjær 2260 sound level meter are used for the sound measurements. The sound level meter is a Class I instrument, with accuracy of ± 1 dB (± 0.5 dB between 800 Hz and 5000 Hz in third-octave bands). The reproducibility of the reported results is within ± 1.5 dB.

The 1.7 m long splitter is placed in the center of the test duct, in Fig. 6 a sketch of an example configuration is shown.

The following shortcomings and limitations have to be taken into account in the analysis of the measurement results:

- at 40 Hz and at 125 Hz system resonances can be detected independently of the flow speed and the test section being empty or not;

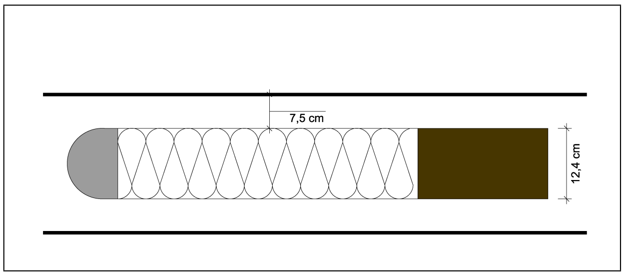

Figure 6. Configuration of the splitter in the test duct.

- the noise due to the flow generation system, even if well reduced by a series of three silencers directly downstream the fan and by one silencer at the air intake, is detectable up to 250 Hz;
- at high frequencies, starting at 6300 Hz, the background noise is not controllable.

3. Measurements

3.1. Tested cases and data processing

Measurements have been performed at four different air bulk velocities V_{ref} derived from the flow speed measured on the axis of the circular duct and with the flow rate kept as constant as possible (± 0.2 m/s) during each measurement series. The accuracy of the air speed sensor is ±1% of range, ± 3% of the measured value. Table III reports these bulk velocities as well as the corresponding V_{gap} in the gap between splitter and test section wall. Different splitters (ranging from full absorbing to *non-absorbing* splitter side walls) have been tested, only tests with *non-absorbing* splitters (steel plate between perforated cover and absorbing cushion, see Section 2.1) are reported here. Tests focus on the effect of different trailing edge extensions and on the influence of covering the perfo-

Table III. Bulk velocities in the circular duct V_{ref} and in the gap between splitter and test section side wall V_{gap}.

		V_{ref}	V_{gap}
v1	[m/s]	9	12.5
v2	[m/s]	11.7	16.2
v3	[m/s]	14.8	20.4
v4	[m/s]	17.5	24.2

rations of the outer plates to create an even (non-perforated) splitter surface. All the sound power level (PWL) values reported here are obtained via subtraction of a reference PWL - that is obtained from measurements with empty test duct (no splitter) - from the PWL obtained with the tested splitter installed in the test section. An additional correction is made for the insertion loss due to the splitter itself. The overall aerodynamic resistance in the empty duct and the one equipped with splitter differs, leading to different loading of the blower; the noise of the air blower has been tested at different loads, and it can be considered constant for each tested velocity at the different pressure loads involved. Table IV reports the pressure drops for different configurations and the average (through the different air speeds) of the pressure drop coefficients defined as:

$$\zeta = \frac{\Delta p}{1/2 \rho V_{ref}^2}. \quad (1)$$

3.2. Sound Power Level comparison

Qualitatively comparable results are obtained for absorbing and *non-absorbing* splitters. Fig. 7 shows the calculated PWL for three different *non-absorbing* configurations:

- tapered wooden extension, with perforation;
- blunt wooden extension, with perforation;
- tapered wooden extension, with no perforation.

Spectra exhibit distinct peaks at frequencies of 400 Hz, 630 Hz and 1000 Hz that do not change with flow speed, indicating that they should be interpreted as a property of the test facility not directly related to self-noise. In the case with perforation significantly higher PWL levels are found in the range above 800 Hz. Spectra exhibit a slight secondary peak at frequencies corresponding to a Strouhal number based on the hole size of the metal perforates as:

$$St_{perf} = \frac{f_{pk} d_H}{V_{gap}} \quad (2)$$

where f_{pk} is the peak frequency and d_H is the diameter of the single perforation hole. The observed value $St_{perf} = 0.3$ is in accordance with previous findings [1]. Tests with the two different wooden extensions (blunt and tapered) do not show significant differences, except for the low frequency range (up to 400 Hz), where the tapered trailing edge shows lower PWL levels in comparison to the blunt one. It is important to remember that the overall pressure drop induced by a tapered trailing edge is lower than for a blunt one, see table IV. The effect of tapering of the trailing edge on the noise spectrum remains similar for the situation with perforated plates as in the case with the perforates covered, even if not shown in the present paper.

Finally, regarding the serrated extensions, no evident difference is measured between the several configurations and the simply blunt one, so no result is shown here.

3.3. Nelson-Morfey scaling

In 1981 Nelson and Morfey [2], through an experimental investigation, established a relation between the aerodynamic sound produced from a series of flow spoilers (orifices) in a rectangular duct and the fluctuation of the drag forces. Moreover they assumed a direct proportionality between the fluctuating forces and the steady state drag force: within the proportional frequency band defined by $(f_c/\alpha,\ f_c\alpha)$ (where f_c is the centre frequency) the ratio of the root mean square force $(F_D)_{rms}$ and the steady state force $\overline{F}_D$ is expected to depend only on the Strouhal number.

$$(F_D)_{rms} = K(St)\overline{F}_D; \quad \text{(band } f_c/\alpha \text{ - } f_c\alpha) \quad (3)$$

Nelson and Morfey suggest a scaling law - that has been formulated by Oldham and Ukpoho [3] also for circular ducts - that is based on the open area ratio σ (defined as the ratio between the constricted and the free areas A_c/A), the constriction velocity V_{gap} and the drag coefficient

$$C_D = \frac{\Delta p}{\frac{1}{2}\rho V_{gap}^2 \sigma^2 (1-\sigma)} \quad (4)$$

In this context, the case under study presents a peculiarity: while the source region (test section) is in a rectangular duct, the transmission is through a circular duct. Accordingly, the scaling laws presented in references [2] and [3] have been combined as shown below, to make them applicable to the examined case:

$$120 + 20\log_{10} K(St) = PWL - 10\log_{10}\left\{\frac{\rho_0 A[\sigma^2(1-\sigma)]^2 C_D^2 V_{gap}^4}{16 c_0}\right\} \quad (5)$$

for f_c below the cut-on frequency, while for f_c above the cut-on frequency:

$$120 + 20\log_{10} K(St) = PWL - 10\log_{10}\left\{\frac{\rho_0 \pi A^2 (St)^2 [\sigma^2(1-\sigma)]^2 C_D^2 V_{gap}^6}{24 c_0^3 d^2}\right\} - 10\log_{10}\left[1 + \frac{3c_0}{8 r f_c}\right] \quad (6)$$

Table IV. Pressure drops and average pressure drop coefficients.

V	Δp Empty duct [Pa]	Δp Blunt, no perf. [Pa]	Δp Tapered, no perf.[Pa]	Δp Tapered, perf. [Pa]
v1	14	62	52	60
v2	27	112	90	100
v3	44	180	150	168
v4	62	250	210	240
ζ_{Average} [-]	0.34	1.41	1.17	1.32

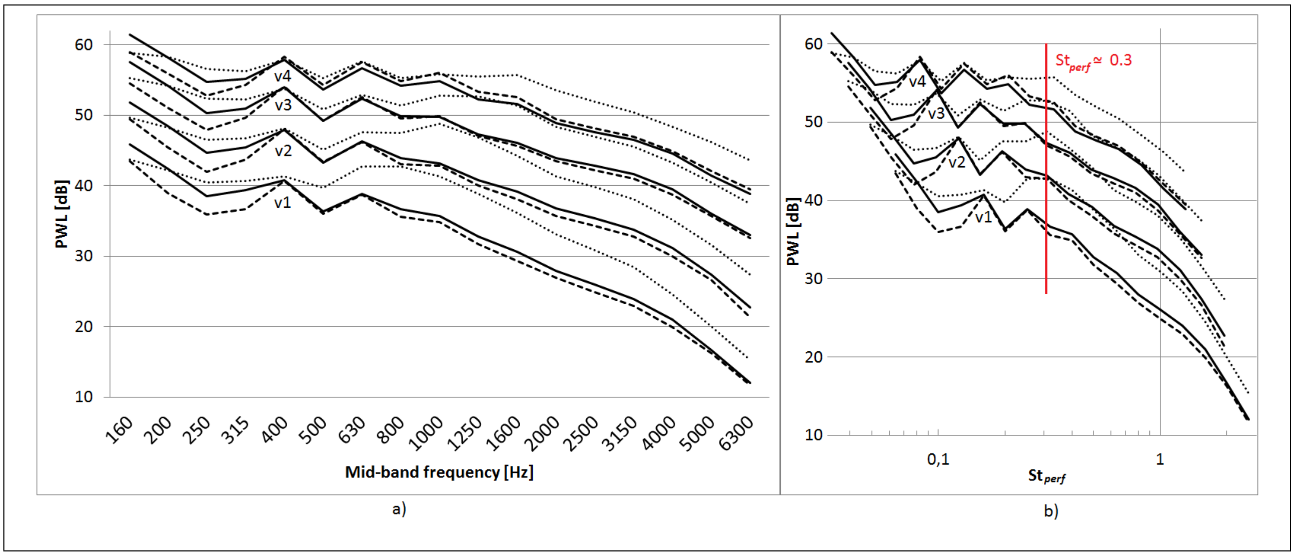

Figure 7. PWL obtained for different *non-absorbing* splitter configurations, plot in function of frequency (a) and $\mathrm{St}_{\mathrm{perf}}$ (b). —— blunt extension, with no perforation, - - - - tapered extension, with no perforation, ······· tapered extension, with perforation.

where d is equal to the obstacle width, c_0 is the speed of sound, r is the duct radius and $St = f_c d/V_{gap}$.

Karekull, Efraimsson and Abom in [4] introduce the concept of average duct resistance, in order to evaluate the contribution of the propagating modes as a function of wave number $k = 2\pi f/c_0$ and duct dimensions. This average duct resistance is:

$$R_{Plane\ wave} = \frac{\rho_0 A}{8c_0}; \quad k < k_{Cut-On} \tag{7}$$

$$R_{Circ.\ d.} = \frac{\rho_0 A^2 k^2 (1 + \frac{3\pi}{4rk})}{48\pi c_0}; \quad k > k_{Cut-On} \tag{8}$$

where r is the duct radius. Compared to the plane wave regime, equation 8 introduces a factor that for high frequencies converges to a component radiating as a dipole in the free field.

Figures 9 and 10 show results that are obtained when the above scaling laws are applied to the PWL data from measurements with the following two configurations:

- a 92 mm width spoiler (thickness 20 mm, height 600 mm) placed in the center of the test section (Fig. 8);
- a non absorbing splitter with perforated surface (same as in Fig. 7).

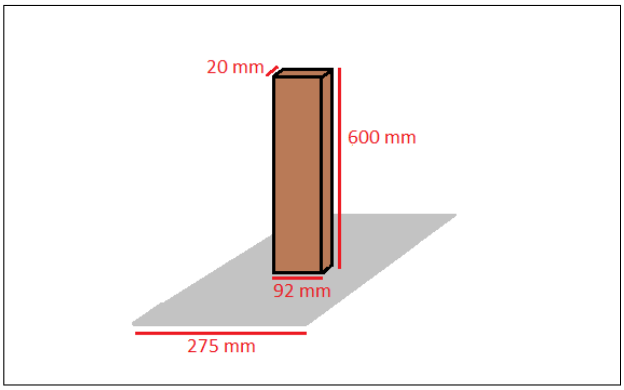

Figure 8. Sketch of the spoiler placed in the center of the test duct.

It is seen that the scaled data collapse well for all velocities tested, especially for the frequency above the cut-on. Karekull, Efraimsson and Abom [4] determine a universal slope for a generic geometry of about $-28log(St)$. This dependency is not clearly found in the results shown in figures 9 and 10, where $-28log(St)$ and $-45log(St)$ slopes are drawn for comparison. The reason of this mismatch, more evident for higher Strouhal numbers, is unknown at the present, and maybe can be referred to the particular test stand. Regarding the slope of the spectra at high fre-

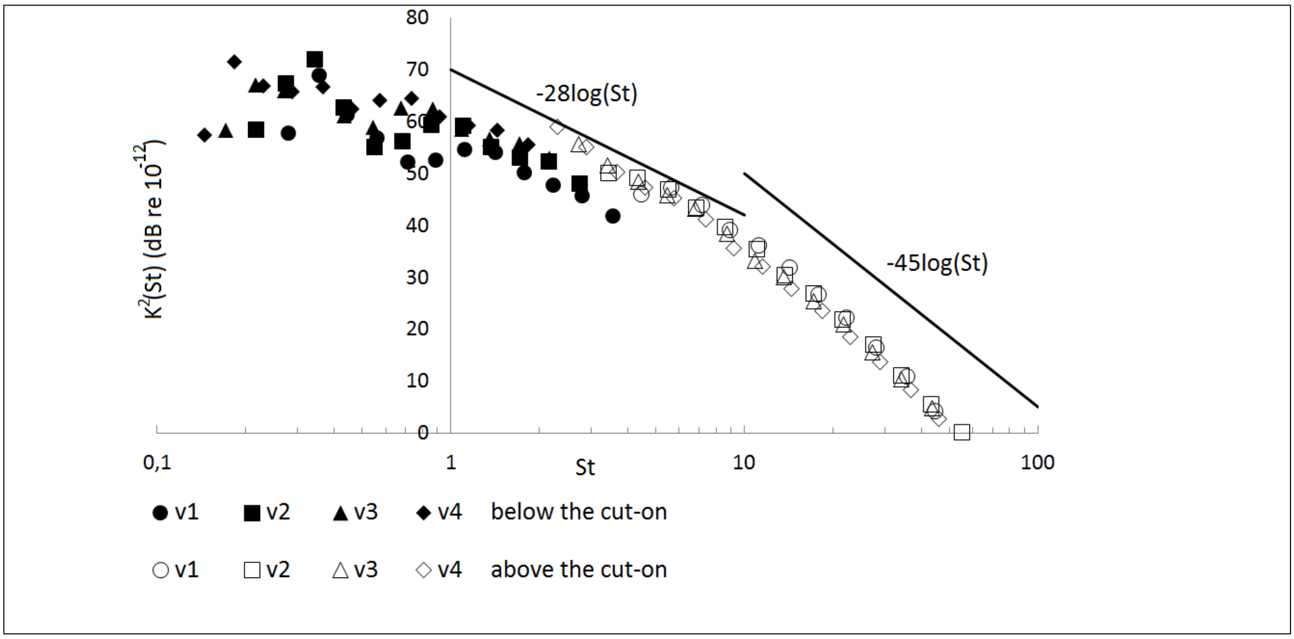

Figure 9. Nelson and Morfey modified scaling for a 92 mm spoiler.

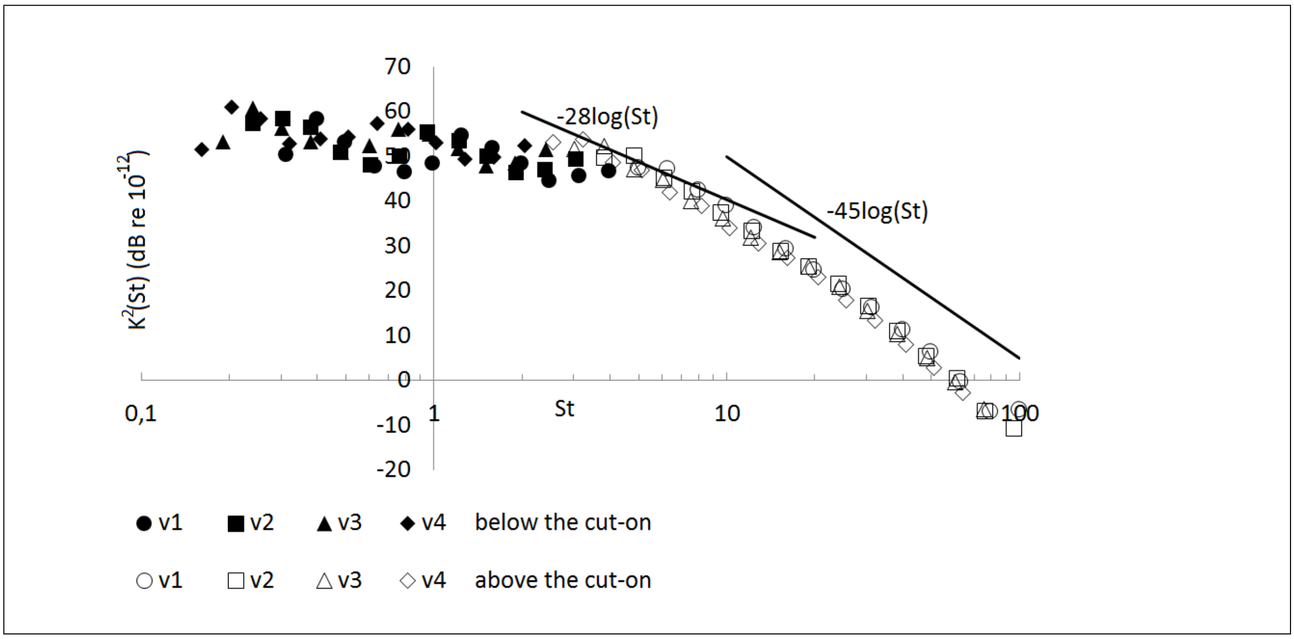

Figure 10. Nelson and Morfey modified scaling for a *non-absorbing* splitter with perforated surface.

quency a considerable scatter is evident in the data presented in [4].

Acknowledgement

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