Experimental and Numerical Characterisation of a Non-Locally Reacting Liner

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Introduction

Noise reduction for various applications is still mainly achieved by the extensive use of passive acoustic wall treatments, so called *liners*. In the field of aircraft noise reduction special emphasis is put on the capability to provide high sound attenuation by utilizing lightweight but also stiff and stable structures, e.g. consisting of Titanium or Nickel alloys. The latter are mainly needed for parts additionally affected by hot and corrosive exhaust gases, e.g. in the exhaust duct of an auxiliary power unit (APU) or the hot stream parts of the aero-engine. This paper examines a recently developed liner structure which combines the mentioned properties of a stable structure and a heat resistant surface. Thus, being applicable for both the aircraft engine nacelle or in exhaust ducts, e.g. of the APU [7]. Due to the patented production process this type of liner is non-locally reacting. The related acoustic properties in terms of energy dissipation, calculated from microphone measurements with grazing flow and impedance are presented. Furthermore, it is shown that the proven techniques for locally reacting liners are even suitable to characterise this special type of liner.

Acoustic Test Rig

The acoustic measurements with grazing flow are performed at the German Aerospace Center (DLR) with a flow tube designed for acoustic testing [1, 2]. The test rig consists of two symmetric duct sections with a square cross-section of 80 mm $\times$ 80 mm (Fig. 1). It is equipped with anechoic terminations at both ends to minimise end reflections. The set-up can be supplied with a main flow up to an average Mach number of 0.27. On top of each section five axial positions for wall-flush mounted microphones are available. They are equipped with G.R.A.S. condenser microphones type 40BP-S1 and type 26AC pre-amplifiers providing an OROS OR36 data acquisition system with the pressure signals. A Monacor KU-516 loudspeaker is connected to the duct cover plate at the end of both sections. The speakers are supplied with signals from the OR36 system, amplified by a Dynacord L300. A mounting with the test object is integrated between the two sections, embedding the sample as the bottom wall.

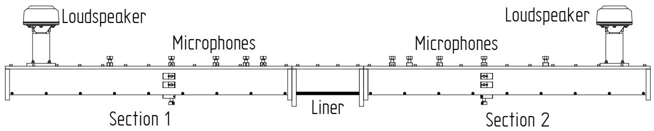

Figure 1: Drawing of the test section (without anechoic terminations).

Test Object

The investigated test object is a (prototype) liner segment with a length of 220 mm and 110 mm in width. Since the surface is covered with a mesh structure the damping performance should not be dependent on flow velocity. Therefore, the liner can be classified as a *linear liner* [11]. This type could be mounted in the intake or bypass duct of aircraft engines, especially where noise of fan and compressor stages have to be reduced. It is manufactured by combining a cell structure of assembled metal strips with a perforated face sheet and a solid backing plate by using a point welding process [8]. The structure is completed with an additional face sheet layer of meshed metal baffle [7]. The connection of the entire structure is made by a new type of contact welding procedure, in which evenly distributed welding spots are build at the upper and lower sides of the strips (see Fig. 2). Based

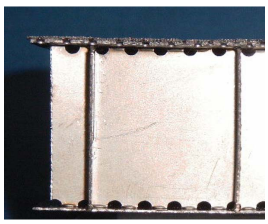

Figure 2: Detail view of the inner structure of the test object, noticeable the welding spots and slots to the neighbouring cells, picture taken from [8]

on this manufacturing process the damper is non-locally reacting because inter-cellular communication is possible. Therefore, the wall impedance (Z) at one position is not only dependent on the frequency, but also affected by the sound pressure and the acoustic particle velocity at other positions of the wall [12]. To prevent fluid exchange with the ambient air the cells at the periphery of the sample were closed using sealant (Fig. 3). This reduces, however, the acoustically effective area of the sample to an axial length (L) of 200 mm and 100 mm in width.

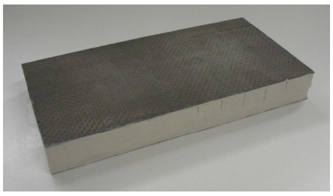

Figure 3: Testobject with sealed lateral cells and meshed metal baffle coating.

Test Configurations

The microphone measurements were conducted for excitation frequencies (f) in the range between 210 and 2060 Hz with frequency steps of 51 Hz. This corresponds to reduced frequencies kD between 0.31 and 3.02, investigating plane wave propagation. Since the field of application of this liner type is related with high flow Mach numbers and only limited installation space, two main parameters are investigated:

- The Mach number is varied for $M = 0,\ 0.1$, and 0.2.
- The normalised active length of the liner L/D is changed between 2.5 and 1.25 by simply turning the test object crosswise, for a constant duct height of $D = 80$ mm.

Measurement Results

The microphone signals were utilised to determine the reflection and transmission coefficients r and t in terms of pressure amplitude ratios of the downstream and upstream propagating waves, according to [1] and [2]. Along with this theory the energy dissipation coefficient Δ depending on the flow Mach number is defined as

$$\Delta^{\pm} = 1 - \left(\frac{(1 \mp M)^2}{(1 \pm M)^2} \cdot \left| r^{\pm} \right|^2 + \left| t^{\pm} \right|^2 \right). \qquad (1)$$

The results are shown in Fig. 4 for downstream (+) and in Fig. 5 for upstream (−) direction.

Grazing flow

This type of liner has a broadband damping performance with a maximum dissipation coefficient around $\Delta^{\pm} = 1$ for $M = 0$. When increasing the Mach number a decreasing dissipation for high frequencies ($f > 1$ kHz) in downstream direction (+) is observed. Furthermore, broadening for lower frequencies and saturation for high frequencies in upstream direction (−) become visible [2]. A maximum is found for both cases, enclosing frequencies between 1.5 and 2 kHz.

Liner length

A different behaviour is found when decreasing the effective liner length to $L/D = 1.25$. For the no flow case this is shown in Fig. 4 and 5. For both directions of propagation and frequencies below 600 Hz the dissipation coefficient has a local maximum. For higher frequencies the curve increases but does not reach the values of the baseline ($L/D = 2.5$). Thus, a non-linear dependency of dissipation coefficient and liner length is indicated. The shape of the curves for upstream and downstream direction is similar, except for a local maximum around 1.2 kHz for upstream propagation. This requires additional analysis.

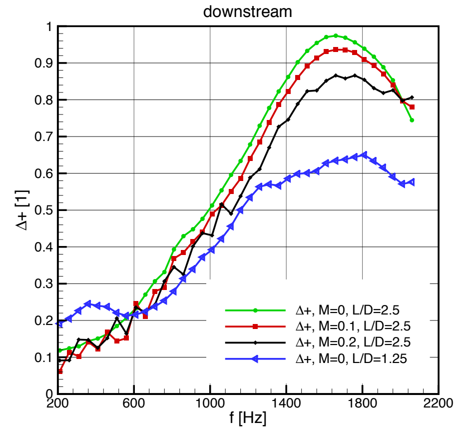

Figure 4: Dissipation coefficients for downstream direction depending on the Mach number M and normalised liner length L/D.

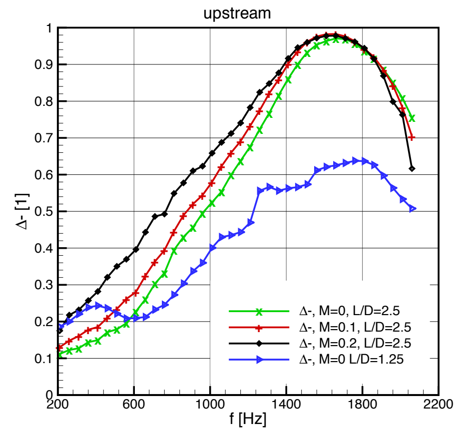

Figure 5: Dissipation coefficients for upstream direction depending on the Mach number M and normalised liner length L/D.

Impedance Eduction

Numerical Model

A time domain CAA method is employed for the simultaneous broadband eduction of the impedance from the measured reflection and transmission data [2]. The method is based on an optimised finite difference discretisation (DRP) for spatial and an optimised Runge-Kutta scheme (LDDRK) for temporal discretisation, both of fourth-order. The time stepping is implemented in 2N storage form. The impedance of the lined duct wall, given by the Extended Helmholtz Resonator model (EHR) according to Rienstra [10], is defined as follows:

$$Z(i\omega) = R_f + i\omega m - i\beta \cot(0.5\,\omega\,T_l - i\,0.5\,\varepsilon). \qquad (2)$$

The five model parameters are the face sheet resistance R_f, the face sheet reactance m, the cavity reactance β, the resistance parameter for the cavity fluid ε as well as the response time T_l. The model describes a locally reacting single degree of freedom liner. Unlike the classical mass-spring-damper analogies it is not limited to low frequencies since the wave propagation inside the cavity is modelled continually. The boundary condition of Myers [6] is used to model the shear layer at the lined surface.
Due to the non-locally reacting structure of the investigated liner sample, the non-local effect on the resulting average panel impedance is studied. This is intended as a first step for the modelling of non-locally reacting liners. By minimising the least squares of the deviation between simulated and measured reflection and transmission coefficients the optimum parameter set for Eq. (2) is found [2]. The MATLAB optimization toolbox is utilised for that. Calculation of the objective function and the Fourier analysis of the time series from the numerical simulation is also performed with MATLAB. The iteration loop takes about one day with up to 300 calls of the CAA method with modified impedance parameters.

Eduction Results

Figure 6 shows the frequency response of the educed impedance for all investigated flow Mach numbers. In the upper part the real part of the dimensionless wall impedance is shown. At the bottom the imaginary part is illustrated. Since the EHR-impedance model allows an extrapolation the shown frequency range is a bit larger than the measured one. However, the extrapolation is not yet validated with experimental data.
For the specific liner length $L/D = 2.5$ the impedance curves are very similar for the investigated Mach numbers. The resistance $R/(\rho\,c)$ decreases from 5 at 250 Hz to around 1 at 1 kHz. The reactance $X/(\rho\,c)$ shows the typical increase from values around -8, crossing zero, and afterwards positive values. The resonance frequency of this damper configuration, in other words the zero of the reactance curve is approximately between 2.4 and 2.5 kHz. An exception is the curve of $M = 0.2$ with a resonance at 2 kHz. The educed resonance frequencies are slightly outside the range being measured. But since the extrapolation is believed to be suitable, those are assumed to be reliable. However, the resonances do not fully match the frequencies for maximum dissipation (Fig. 4 and 5) since an optimum impedance can also include an imaginary part [4].
Real and imaginary part of the impedance change only little with increasing Mach number. A minor flow influence is visible only in a change of the resistance for frequencies lower than 1 kHz. The reactance is virtually unaffected by the flow. The changing propagation conditions with grazing flow are taken into account with the impedance model in combination with the boundary condition of Myers [6]. Therefore, the observed change of the impedance corresponds to a change in the characteristics of the liner through the flow, although the effect is very small for the investigated liner.
When changing the liner length the impedance curves also shift to slightly higher resistance values, especially for frequencies above 1.5 kHz. Below this frequency the curve is flattened compared to the baseline curve of $M = 0$ and $L/D = 2.5$. In contrast the reactance curve shows nearly no deviation for frequencies above 1.5 kHz, whereas the asymptotic trend for lower frequencies is smoothed to just a small dent. The EHR-model parameter of the internal damping or cavity resistance of the resonator cells ε in this case provides a significant contribution to the educed impedance (see Eq. 2). Therefore, the liner behaves similarly to dampers, filled with absorbent materials like foam.

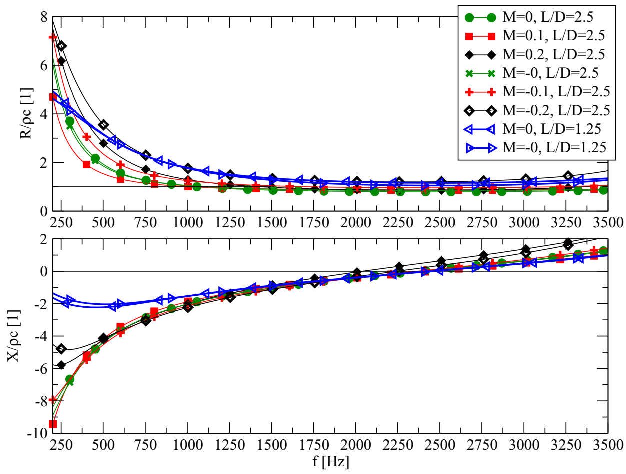

Figure 6: Impedance eduction results, considering the influence of the flow Mach number and direction of wave propagation as well as liner length to duct height ratio (L/D). Both curves (normalised reactance R and resistance X) are extrapolated for frequencies exceeding 2 kHz.

Conclusion

It is shown that the described techniques commonly utilised for locally reacting liners are also suitable to study non-locally reacting liners. A liner sample was characterised according its acoustic damping performance using the approved measurement techniques and the formulation of the energy dissipation coefficient along with a numerical impedance eduction method based on the Extended Helmholtz Resonator (EHR) model. Some properties are observed and considerations are made:

- Compared to Eaton [3] and Murray [5] which investigated the influence of drainage slots at the bottom of nacelle acoustic liner cells, the studied liner provides inter-cellular communication in the upper and lower part of the cells. This might cause additional changes of cell volume properties due to the fluid interchange caused by grazing flow and acoustic excitation.
- The calculated dissipation is per definition dependent on the flow Mach number (see Eq. 1), which

represents a sort of acoustic energy transport with the grazing flow [2].

- The maximum dissipation for upstream and downstream direction is found for a frequency range enclosing typical blade passing frequencies of aero-engine fans.
- The sample behaves as linear liner, since its impedance is altered only for frequencies below 1 kHz when varying the flow Mach number. Moreover, mainly the resistance is affected.
- The liner length is important for both the dissipation values and educed impedance. Since the impedance curve does not change significantly for frequencies above 1 kHz at the no flow case, the presented model approach is valid for this liner type.
- The internal damping in terms of the cavity resistance ε increases for $L/D = 1.25$. Presumably, this is caused by the losses due to communication of the individual cells. Such a non-local effect, however, is not fully defined with a single average wall impedance. Modelling the wall as a damped membrane could take into account waves which enter the liner at a certain point and leave it at another. Nevertheless, a refinement of the model is needed to describe those effects in a more physical way.

Acknowledgments

This research is funded by the German Research Foundation (DFG) within the LIMiT project. The authors would like to thank Dr.-Ing. Ulf Michel and Dipl.-Ing. Claus Heuwinkel for the support and the fruitful discussions on non-locally reacting liners. Furthermore, the assistance of the students Benjamin Bock and Daniel Löchert during the measurements is highly appreciated.

Nomenclature

c	Speed of sound
CAA	Computational Aero-Acoustics
D	Duct height
DRP	Dispersion Relation Preserving (scheme)
EHR	Extended Helmholtz Resonator (model)
f	Frequency
kD	Reduced frequency/wavenumber
i	Imaginary unit
L	Effective liner length
L/D	Ratio of liner length and duct height
LDDRK	Low Dissipation Low Dispersion Runge Kutta (scheme)
m	Facesheet reactance (EHR model)
M	Mean Mach number in the duct
r	Reflection (amplitude ratio)
R	Resistive part of the impedance $R = \mathrm{Re}\{Z\}$
R_f	Facesheet resistance (EHR model)
t	Transmission (amplitude ratio)
T_l	Time delay (EHR model)
X	Reactive part of the impedance $X = \mathrm{Im}\{Z\}$
Z	Complex impedance
β	Cavity reactance (EHR model)
Δ	Energy dissipation
ε	Cavity resistance (EHR model)
ρc	Characteristic impedance of the fluid
ω	Angular frequency

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