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SOUND PROPAGATION IN NARROW CHANNELS WITH ARBITRARY CROSS SECTIONS AND SUPERIMPOSED MEAN FLOW WITH APPLICATION TO CHARGE AIR COOLERS

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Abstract

Charge air coolers are used on turbo charged IC-engines to enhance the overall gas exchange performance. The cooling of the charged air results in higher density and thus volumetric efficiency. Important for petrol engines is also that the knock margin increases with reduced charge air temperature. A property that is still not very well investigated is the sound transmission through a charge air cooler. The pressure drop in the narrow cooling tubes results in frequency dependent resistive effects on the transmitted sound that is non negligible. As the cross sections of the cooling tubes are neither circular nor rectangular, no analytical solution accounting for a superimposed mean flow exists. The cross dimensions of the connecting tanks, located on each side of the cooling tubes, are large compared to the wavelength for engine breathing noise, here including frequencies up 1.5 kHz, so three dimensional effects are important. In this study an acoustic two-port for sound propagation in narrow tubes, including the effect of viscous and thermal boundary layers, is calculated utilizing a 2D finite element solution scheme. Analytical solutions for circular cross sections are additionally calculated for comparison. The two-port is thereafter combined with 3D acoustic finite element modelling to represent the transmission properties of the charged air when passing the complete air-to-air charge air cooler. From this a linear frequency domain model for the entire charge air cooler is extracted in the form of a two-port. The frequency dependent transmission loss is calculated and compared with corresponding experimental data. Finally, there is a discussion of the results and the potential of using charge air coolers to control the acoustic response of intake systems.

1. INTRODUCTION

The recent trend of downsizing internal combustion (IC) engines and using turbochargers to maintain the engine torque and power imposes some additional noise phenomena not created by naturally aspirated engines. The frequencies of these sources of noise are mainly of higher frequencies [1] than those produced by engine breathing. Many turbocharged engines are equipped with charge air coolers (CAC); a device intended to increase the overall performance of the engine. The cooling of the charged air results in higher density and thus volumetric efficiency. Important for petrol engines is also that the knock margin increases with reduced temperature.

The parameters of main interest when designing a charge air cooler are normally the pressure drop and the heat exchange efficiency. According to the literature survey in [2] there are predictive models available describing the thermal efficiency [3], and also models treating flow unsteadiness [4, 5 and 6]. Still they are only evaluated in terms of heat transfer performance, pressure drop and gas-exchange properties mainly affected by lower frequencies. However, what seem to have been overseen are the acoustic properties which are still not very well investigated. To the authors' knowledge the sound attenuation properties are only dealt with in three publications [2, 7 and 8]; where [2] and [7] are making use of 1D models based on acoustic two-ports to assemble a complete model for a charge air cooler. In [8] the present authors describe a hybrid model where acoustic finite elements are coupled to acoustic two-ports calculated using the classical Kirchhoff solution for sound propagation in narrow circular ducts.

In this paper the linear frequency domain hybrid model in Reference [8], based on 3D acoustic finite elements and two-ports, is improved in order to account for narrow non-circular cross-sections using a 2D finite element solution scheme. This approach is shown to generate good results when predicting frequency dependent transmission loss up to 1500 Hz without adjustment of any input geometrical dimension. The suggested technique thereby offers a new possibility to tune the acoustic properties of charge air coolers.

2. MODELLING

An air-to-air charge air cooler consists basically of three types of elements; the cooling tubes where the heat exchange process occurs, the inlet- and outlet tanks connecting the charge air ducts to the cooling tubes and the cooling fins outside the cooling tubes. Assuming a 1D acoustic state throughout the charge air cooler, the sound propagation can be described using acoustic two-ports (or four-poles). This will be consistent as long as the highest frequency of interest stays well below the cut-on frequency for the first non plane wave. The frequency domain relationship between the acoustic states at sections representing the inlet and outlet of a two-port can be written

$$\begin{bmatrix} P_{in} \\ Q_{in} \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} P_{out} \\ Q_{out} \end{bmatrix} \tag{1}$$

where P denotes the plane wave acoustic pressure and Q the acoustic volume velocity. For simple generic geometries, such as circular ducts, it is possible to obtain the complex valued components in the transfer-matrix from analytical expressions. For a component built up of three sub-components coupled in series, such as the charge air cooler, the assembled two-port can be calculated from

$$\mathbf{T}_{TOTAL} = \prod_{n=1}^{3} \mathbf{T}_n \tag{2}$$

The narrow cooling tubes only carry plane waves in a wide frequency range; in the cases studied here up to several kHz. They are therefore ideal to be described by two-ports. The cross-dimensions of the inlet- and outlet tanks of typical charge air coolers are much larger than the cross-dimensions of the cooling tubes. Therefore 3D effects can become important well below 1.5 kHz, as will be demonstrated by the case studied here. To include these 3D effects a hybrid finite element/two-port methodology is developed. In this coupled approach the inlet/outlet tanks are modelled by 3D acoustic finite elements and the cooling ducts by two-ports. The coupling is performed using the concept of generalized admittance matrices in the commercial software LMS/Sysnoise [9]. To simulate the effect of yielding walls and propagation losses in the inlet and outlet sections, made of plastic, damping is applied as a complex speed of sound. Based on the investigation in Reference [10], an engineering estimate for the damping in intake system components made of plastic is obtained by putting the imaginary part of the speed of sound equal to one percent of the real part. This methodology gives the opportunity to model charge air coolers accurately up to the cut-on frequency for the first plane wave in the main duct. Above this frequency higher order multi-port models are required or if only results in frequency bands are of interest the use of power based (SEA type) models can yield good results. In this study the cut-on frequency for the main duct (φ = 66 mm) is about 3 kHz. However, the upper frequency limit was chosen to 1.5 kHz. In order to verify the methodology above this frequency additional measurements are required.

Since the shortest dimension of the cross section of a sub-channel in a cooling tube is in the order of 1 to 3 mm, the sound transmission will be influenced by the viscous and thermal boundary layers. A number of models, normally expressed in the form of a two-port, have been presented for the acoustics in such devices. Most of these studies are devoted to the acoustics of catalytic converters, but since the dimensions are of the same order of magnitude the models will apply also for charge air coolers. One early model for sound propagation where boundary layers are of importance is the classical Kirchhoff solution. A number of improved models have been presented [11-16] where the effects of a superimposed mean flow is additionally treated. In the paper by Astley & Cummings [13] a FE solution scheme is described, enabling solutions for arbitrary cross-sections where a mean flow is present. This solution scheme, where the walls are assumed being rigid and impervious and the mean flow is assumed being laminar, incompressible, fully developed and parallel, is the base for the present work.

The fundamental linearized equations for a viscothermal fluid, as formulated by Zwikker and Kosten, describing wave propagation in the x-direction, assuming an ideal gas and an incompressible mean flow while neglecting dissipation, are for harmonic time variation, see Reference [12]:

$$\int_R \left(\left(i\omega + U_0 \frac{\partial}{\partial x} \right) \rho' + \rho_0 \frac{\partial u'_x}{\partial x} \right) dR = 0 \,, \;\; \rho_0 \left(i\omega + U_0 \frac{\partial}{\partial x} \right) u'_x = -\frac{\partial p'}{\partial x} + \mu \left(\frac{\partial^2 u'_x}{\partial y^2} + \frac{\partial^2 u'_x}{\partial z^2} \right), \qquad (4, 5)$$

$$\rho_0 C_p \left(i\omega + U_0 \frac{\partial}{\partial x} \right) T' = \left(i\omega + U_0 \frac{\partial}{\partial x} \right) p' + k_{th} \left(\frac{\partial^2 T'}{\partial y^2} + \frac{\partial^2 T'}{\partial z^2} \right), \qquad \frac{p'}{p_0} = \frac{\rho'}{\rho_0} + \frac{T'}{T_0} \,. \qquad (6, 7)$$

Here ρ_0, U_0 and T_0 are mean density, flow velocity and temperature respectively, $\mathbf{u}'$ is the acoustic velocity that comprises u'_x, u'_y and u'_z for each direction, p', ρ' and T' are the acoustic pressure, density and temperature respectively, C_p is the specific heat at constant pressure, k_{th} the thermal conductivity and R is the cross-section of the cooling tube. The boundary conditions that must be fulfilled are $\mathbf{u}' = T' = 0$ on the wall ∂R. The transverse components u'_y and u'_z have been eliminated from the continuity equation by taking the integral over the tube cross-section and applying the boundary conditions. Following Astley and Cummings [13] a harmonic plane wave type of solution to these equations, where the equations have been non-dimensionalized using the mean flow quantities and the adiabatic speed of sound, can be obtained by the following ansatz:

$$u' = c_0 u\left(y^*, z^*\right)\exp\left(i\omega t - ik\lambda x\right), \qquad \rho' = \rho_0 \rho\left(y^*, z^*\right)\exp\left(i\omega t - ik\lambda x\right), \tag{8, 9}$$

$$p' = p_0 p \exp\left(i\omega t - ik\lambda x\right), \qquad T' = T_0 T\left(y^*, z^*\right)\exp\left(i\omega t - ik\lambda x\right). \tag{10, 11}$$

Here $k = \omega/c_0$, λ is a dimensionless wave number and y^* and z^* are non-dimensional co-ordinates, scaled using the half-width of the cross-section. Via a weak (Galerkin) FE formulation an eigenvalue equation for the unknown axial wave number can now be obtained as described in [13] in the following form:

$$\left[A - \lambda B\right] = \begin{bmatrix} p \\ \mathbf{u} \\ \mathbf{T} \end{bmatrix} = \begin{bmatrix} 0 \\ \mathbf{0} \\ \mathbf{0} \end{bmatrix}, \tag{12}$$

where **u** and **T** are the vectors describing the mode shape for the velocity and temperature field. In the absence of flow only two non-trivial roots will appear where one is minus the other, describing wave propagation in the positive and negative axial direction. When mean flow is present a full set of non-trivial eigenvalues exists whereof two will be similar in character as the no-flow solutions. Finally to obtain the two-port matrix the velocity, given by the corresponding eigenvector, has to be averaged over the cross-section in order to calculate the wave impedance.

3. MEASUREMENTS

The charge air cooler used for validation of the model is taken from a passenger car in series production and is of air-to-air type. A picture of the cooler and the inner dimensions of the cooling tubes are shown in Figure 1. The cooler is of brick type with a relatively small height and contains ten cooling tubes. To increase the heat exchange efficiency a folded metal sheet is mounted inside the cooling tubes, dividing the tubes into almost triangular sub-channels with the length of the shortest side in the order of 1 to 3 mm.

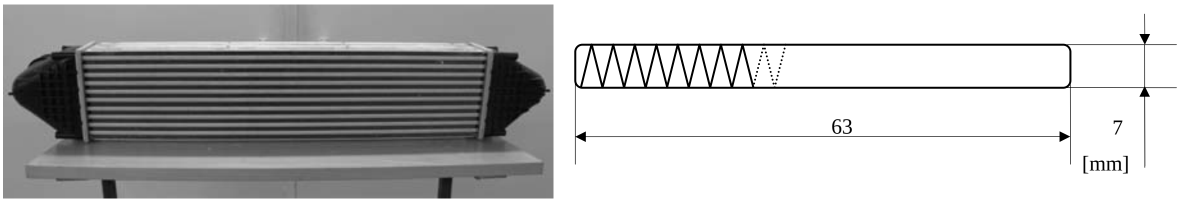

Figure 1. Charge air cooler for passenger car. Dimensions represent the interior of one cooling tube.

The measurements have been performed in order to validate the proposed model, but also to establish the acoustic performance of the charge air cooler at cold conditions. All experiments were performed at room temperature using the flow acoustic test facility available at MWL/KTH see reference [8]. The Mach number in the main duct (φ = 66 mm) was varied between 0 and 0.1; values chosen as being representative for engine operating conditions. This implies that in the cooling tubes, where the area is expanded by a factor 1.2-

1.3, the Mach number will be about 0.08. The test ducts used during the experiments consisted of standard steel pipes, with diameters related to the in- and outlet of the charge air cooler. Eight loudspeakers, equally divided between the up- and downstream side of the rig, were used as acoustic sources. The test rig is terminated at each end by a dissipative silencer to reduce the effects of standing waves. Fluctuating pressures were measured by using six condenser microphones (Brüel and Kjaer) flush mounted in the wall of the steel pipes. The two-port matrix for the test object was obtained using the source switching technique as described in [17]. In order to minimize the effects of flow noise at the microphones, source correlation using the loudspeaker voltage signal was performed.

According to the results shown in Figure (2) there is little effect of flow on transmission loss. At low frequencies the TL will approach a constant level related to the pressure drop of the complete unit. It is known that such an effect can be added to the model by a lumped element two-port defined by this pressure drop [18].

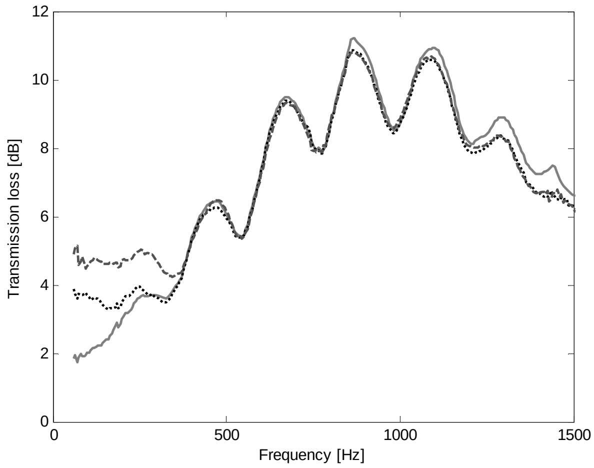

Figure 2. Effect of flow on frequency dependent transmission loss for complete charge air cooler: —, No flow; …, Mach number for mean flow = 0.075; ---, Mach number for mean flow = 0.1.

4. MODEL VALIDATION AT COLD CONDITIONS

Besides studying the hybrid FEM/two-port models, a complete charge air cooler model based entirely on two-ports have been built in order to estimate the need for 3D modelling. Since the two-port models are much faster to build and analyze, the use of finite elements should be limited for use at frequencies where an improvement can be expected.

4.1 Cooling tubes

As the cooling tubes in the test object are split into almost triangular sub-channels, a shape to which analytical solutions does not exist, the FE solution scheme of the equations given in the previous section is used. The accuracy of the solution will depend strongly on the ability of the FE mesh to resolve the boundary layers. For this particular case these turned out to be comparably thin why a rather fine mesh is needed as is shown in Figure (3). In Reference [8] it was shown in that an analytical wave propagation model based on the classical Kirchhoff solution for circular tubes with the diameter equal to the hydraulic diameter gives a good approximation. The difference between the two modelling approaches is illustrated in Figure (4) where the transmission loss for one single sub-channel is shown. As can be observed the difference is small. Concerning the overall acoustic behaviour it is important to note that the damping is small at low frequencies and increases with frequency, indicating that the tubes are more useful as silencers for compressor related noise than the engine breathing noise which consists mainly of lower frequencies.

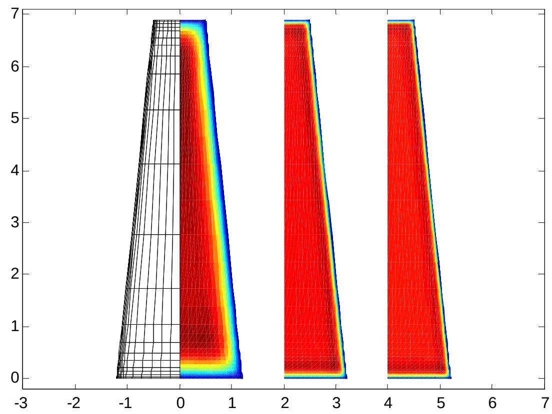

Figure 3. Finite element mesh used for of one sub-channel and the predicted normalized acoustic velocity mode shapes for 50, 500 and 800 Hz (from left to right).

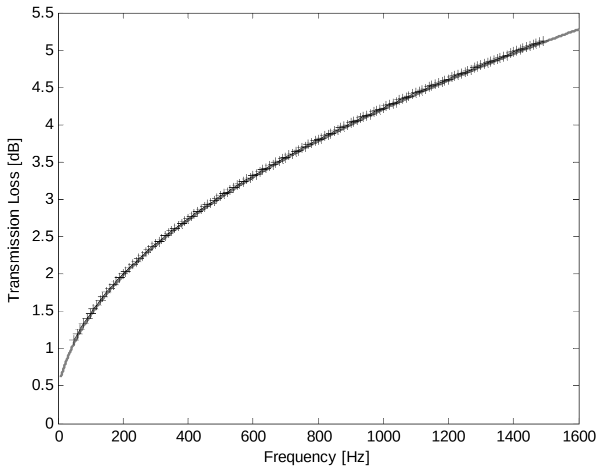

Figure 4. Predicted frequency dependent transmission loss for one capillary tube at no flow conditions: —, Kirchhoff solution for narrow circular cross-section; +++, FE-solution for narrow triangular cross-section.

4.2 Complete charge air cooler

In order to validate the models for the entire CAC, the predicted and measured transmission loss for the no flow case is shown in Figure 5. The hybrid model as well as the two-port model both shows a good agreement. The hybrid model performs best with discrepancies less than 1.5 dB. Important is that the accuracy of the two-port model levels off as the frequency exceeds 1000 Hz. Indeed this is not surprising since this is the cut-on frequency for the largest cross dimension of the inlet- and outlet tanks. This is therefore the expected behaviour as the two-port model is based on the assumption of 1D wave propagation. The sound pressure distribution at 1250 Hz, shown in Figure 8, further verifies this observation. The non-planar wave in the tanks is clearly visible, while the main ducts are still well below the cut-on frequency, verifying the two-port assumption for the complete CAC. Note that the tanks are moved together in the plot and are not representing the ordinary position.

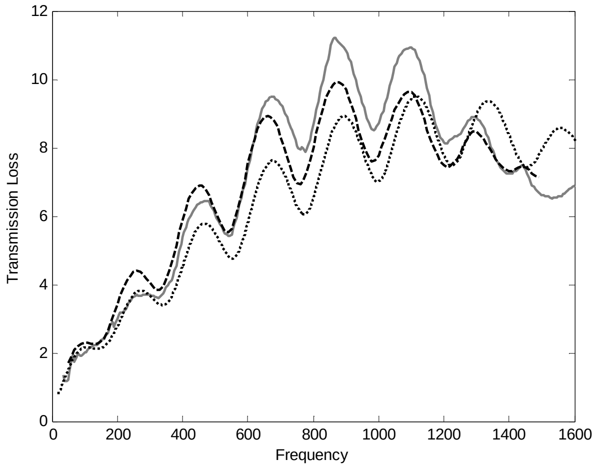

Figure 5. Transmission loss for CAC: ---, Predicted using hybrid FEM / two-port model, …, Predicted using two-port model (SIDLAB [19]) for narrow circular duct; —, Measured.

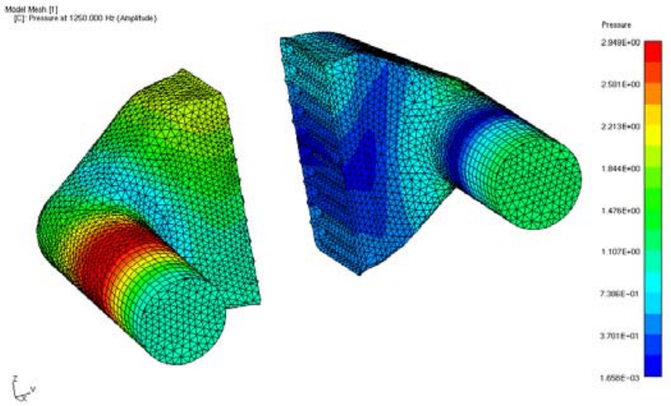

Figure 6. Sound pressure distribution at 1250 Hz in inlet- and outlet tanks obtained using hybrid FEM / two-port model.

5. CONCLUSIONS

Sound transmission through a charge air cooler has been studied for one test case. The frequency range under consideration was low and medium where non-plane waves exist. A hybrid model, based on 3D finite elements and a two-port calculated using 2-D finite elements where viscothermal boundary layers are taken into account, has been developed and validated to experimentally obtained data for transmission loss at room temperature. It is found that from using the proposed hybrid methodology the overall acoustic behaviour can be predicted with good accuracy in the frequency range under consideration (50-1500 Hz).

The air-to-air noise frequency spectra of turbocharged engines consists of low frequencies from engine breathing as well as higher frequencies related to compressor operation (whine, whistle) and transient flow phenomena (sigh). A broadband silencer is therefore a device of great interest for fulfilling the Pass-By noise regulations as well as for creating a desirable sound quality. A charge air cooler contains two of the most widely used sound attenuations measures; the expansion chamber and the narrow resistive duct, and offers thereby possibly underestimated capabilities for broadband noise silencing. As the charge air cooler is installed downstream the compressor, the noise, that is radiated from the compressor and travels upstream towards the intake orifice, is of course not affected by the charge air cooler. However, the compressor noise, that is transmitted downstream towards the intake manifold and thereafter radiated through the duct walls, will be attenuated by the charge air cooler. What is also interesting is that the low frequency engine firing harmonics as well as overhearing through an EGR-system can also potentially be reduced by the charge air cooler. In order to take full advantage of this possibility, the proposed model can be used for optimization studies.

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