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Acoustics of Suction Mufflers in Reciprocating Hermetic Compressors

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ABSTRACT

Optimisation of acoustic properties of suction mufflers inside hermetic compressors has become an important research field. One useful measurable quantity for acoustic optimisation of suction mufflers is the insertion loss. This quantity is strongly dependent on frequency, which is mainly due to the presence of troughs and peaks at certain frequencies. Consequently, this makes it difficult to optimise the insertion loss at all frequencies. In practice, acoustic optimisation must be focused around a limited number of frequencies, for instance, those of certain source components and cavity modes. In this paper some new insights concerning the dependence of the insertion loss on the boundary conditions inside the muffler cavity and around the suction inlet are given. Furthermore, relatively simple experimental methods for the characterisation of sources and cavity modes will be discussed.

1. INTRODUCTION

In recent years there has been a growing demand for energy efficient and less noisy household compressors. Energy optimisation often involves a reduction of the pressure losses in the flow paths of the suction line. A reduction of the losses can in many cases be obtained by increasing the cross-sectional area of the internal tubes of the muffler. This, however, usually increases the sound power radiated from the inlet of the muffler. In order to reduce the noise, which originates from the suction valve movements, the acoustic engineer needs to be able to gain detailed knowledge about the source and how the noise is transmitted through the muffler. This paper will have an emphasis on the proper acoustic characterisation of suction mufflers.

Among the researchers of muffler acoustics there are traditionally three ways to characterise the performance of mufflers. The performance parameters are the transmission loss (TL), noise reduction (NR) and insertion loss (IL), all of which are dimensionless quantities (Munjal, 1987). According to Prasad and Crocker (1981), the latter quantity is the most useful performance parameter. They claim that neither TL nor NR are particularly useful because they are independent of the source characteristics and hence they are unable to exactly predict the changes in the radiated sound power upon design changes. They can only be used as guidance parameters. It must, however, be emphasised that these parameters are still useful when some of the boundary conditions are unknown. In recent studies TL has successfully been used by Lee *et al.* (2002) and Yoshimura *et al.* (2002). In fact, the former research team used a combination of calculated TL and measured IL, which yields additional information about the acoustic performance.

In the automotive industry IL is used extensively to characterise exhaust systems because they are the last component in the acoustic transmission path. This is, however, not the case when considering suction mufflers in hermetic compressors. In this case, a rigorous treatment of the acoustic transmission path must consist of the suction muffler, compressor cavity and compressor shell. It is, in fact, possible to make such coupled acoustic model calculations. However, the assembling of the model is a tedious task and experimental validation of the model is difficult. In particular, the coupling between the compressor cavity and shell is very sensitive with respect to acoustic and structural modal shapes, frequencies and damping. Nevertheless, a detailed study of these interactions has been performed by Buligan *et al.* (2002). In the work presented here, a simplified approach will be adopted, where only the performance of the muffler is studied. In other words, only the insertion loss of the muffler has to be considered in order to predict the acoustic performance of the whole system. In a heavily dampened system, this approach is presumably justified because the radiation impedance around the muffler inlet would have a smooth

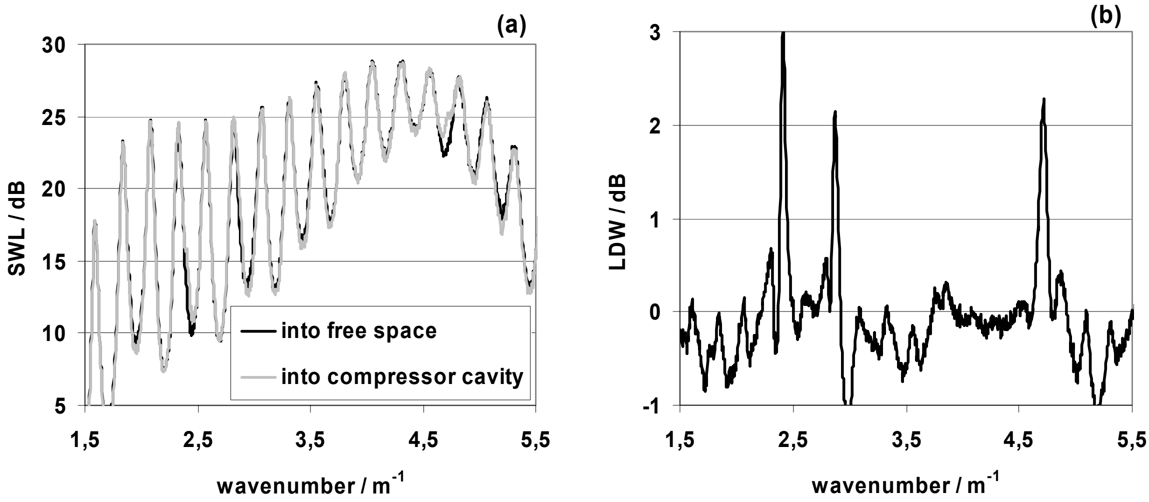

Figure 1: (a) transmitted sound power, SWL, and (b) level difference of transmitted sound powers, LDW.

frequency dependence. This, in turn, ensures that there are no abrupt changes in the transmitted sound power through the whole system.

In the following section, it will be shown experimentally, that the system consisting of the compressor cavity and shell is indeed a heavily dampened system. Furthermore, the measurements can be used for the identification of the most dominant cavity resonances.

2. SOUND POWER DISSIPATION IN THE COMPRESSOR CAVITY

The dissipation of sound power in the compressor cavity has been measured by use of the two-microphone intensity method based on the finite difference approximation (Fahy, 1977). The experimental hardware set-up consists of a 2 m long circular tube, which at one end has a small loudspeaker and at the other end is connected to the suction connector of the compressor. The system contains air rather than cooling medium. This will presumably not affect the forthcoming conclusions because the dissipation within the medium itself is negligible relative to the dissipation in the boundary layer close to the surfaces of the compressor cavity.

The sound intensity in the tube was measured by two probe-microphones situated around half-way between the ends of the tube. Figure 1 (a) shows the transmitted sound power when (*i*) the compressor is hermetically closed and (*ii*) the top cap is removed, this allows the sound waves to escape into free space. Note, that the abscissa in Figure 1 is given in terms of wavenumber (frequency divided by the speed of sound), which enables a comparison between different acoustic media. It is seen, that the two curves are nearly identical, which shows that the dissipation obtained by radiating into the compressor cavity and into free space are nearly the same. The curves differ only around the cavity resonances, where the dissipation inside the compressor cavity is large relative to that into free space. This indicates that the acoustic energy dissipation inside the compressor cavity is relatively high. The level difference between the two curves shows at which wavenumber the cavity resonances are excited: see Figure 1 (b). Consequently, the insertion loss of the muffler should be maximised around these frequencies. Note, that the relative levels around the resonances in Figure 1 (b) do not necessarily give a ranking of the degree to which the cavity resonances can excite the shell. This is because the degree of excitation is dependent on the impedance and pressure magnitude around the suction inlet. Furthermore, it is usually only the three lowest cavity resonances that are able to induce a considerable radiation from the shell. Gavric and Darpas (2002) have elegantly shown that these resonances excite translational motions of the shell, which, in the case of household compressors, are well below its lowest structural modes.

3. SOURCE CHARACTERISATION

The sound power transmitted through the muffler is critically dependent on the source characteristics i.e. source strength and impedance. There are two important mechanisms for the generation of noise during the suction stroke. Firstly, fluttering of the suction valve modulates flow and gives rise to noise of the monopole type. In this case, when doing acoustic model calculations a velocity boundary condition can be put across the suction hole. Secondly, noise is generated due to oscillating pressures on the edge of the suction hole and on the surface of the valve. This generation mechanism is referred to as the dipole type. Howe (1998) has given a comprehensive account of these mechanisms.

At Danfoss some preliminary measurements have been performed in order to determine the source strength of suction valves under realistic operating conditions. The experimental set-up consists of a hermetic closed circuit, which holds the cooling medium. A 30 m long tube with a diameter of 30 mm is directly connected to the suction port. In the tube, about 1 m away from the suction port, a probe microphone is located. The microphone measures the sound pressure of the travelling sound waves, which to a good approximation are travelling in only one direction away from the source. In other words, due to the attenuation in the remaining part of the tube, the source will experience an acoustic load corresponding to an anechoic termination. Under such conditions, the source strength can be calculated straightforwardly from the measured sound pressure. These measurements can be used to detect changes in the source strength upon design changes of the suction valve. A significant complication, however, is the determination of the source impedance. Unfortunately, the source impedance is not measured easily since it involves careful microphone measurements together with strong acoustic drivers, which must exceed the sound pressures generated by the suction valve during normal operating conditions. Ih *et al.* (1998) have performed such measurements on an open system using ambient air.

4. INSERTION LOSS

The experimental validation of acoustic model calculations is an important issue since it gives confidence in the models and enables improvements of the model. Both NR and IL are useful for experimental validation because they are easily measured. Noise reduction is the frequency response function of two microphone signals usually located at the inlet and outlet portions of the muffler. A high signal to noise ratio can be obtained by using probe microphones where the tip of the probe is located inside the piping of the muffler. Insertion loss, on the other hand, is based on sound power rather than sound pressure. The IL is given by

$$IL = \frac{SW_{source}}{SW_{muffler}}, \qquad (1)$$

where SW_{source} is the sound power radiated by the source alone (reference source) and SW_{muffler} is the sound power radiated by the system consisting of source and muffler. Fortunately, in the case of suction mufflers, where the dimension of the suction inlet is small relative to the wavelengths usually considered, the sound pressure in the close vicinity of the object can be used rather than sound power. The sound power radiated from an object can be calculated according to

$$SW = \int_S \frac{1}{2} \mathrm{Re}[p v_n^*] dA = \frac{1}{2} \int_S |p|^2 \, \mathrm{Re}[Z_n] dA, \qquad (2)$$

where S is a virtual surface enclosing the object, v_n and Z_n is the normal components of the velocity and impedance relative to S, respectively. The magnitude of the sound pressure is nearly constant over S and therefore SW can be approximated by

$$SW = C|p|^2, \qquad (3)$$

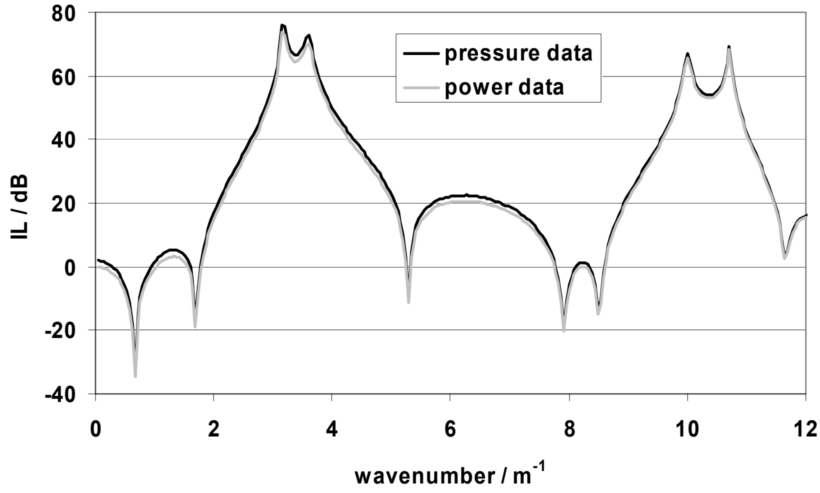

Figure 2: Calculated IL based on either acoustic pressure or power data.

where p is the sound pressure somewhere on S. The factor C in Equation (3) is the same for both source and suction inlet and it is given by

$$C = \frac{1}{2} \int_S \mathrm{Re}[Z_n] dA \,. \tag{4}$$

Thus IL can be measured as the ratio between sound pressures. When doing measurements, the source should be acoustically compact, *e.g.* a small loudspeaker, and the sound pressure should be measured at the same distance from the source and the suction inlet. Figure 2 shows a comparison between the calculated insertion loss using either sound power or pressure on a two-chamber type muffler used in the Danfoss N-type compressors, which have cooling capacities in range 100 – 200 Watts. A boundary element (BE) and a schematic model of the muffler is show in Figure 3. The IL calculation based on pressures used a data recovery point 5 mm away from the source and the suction inlet.

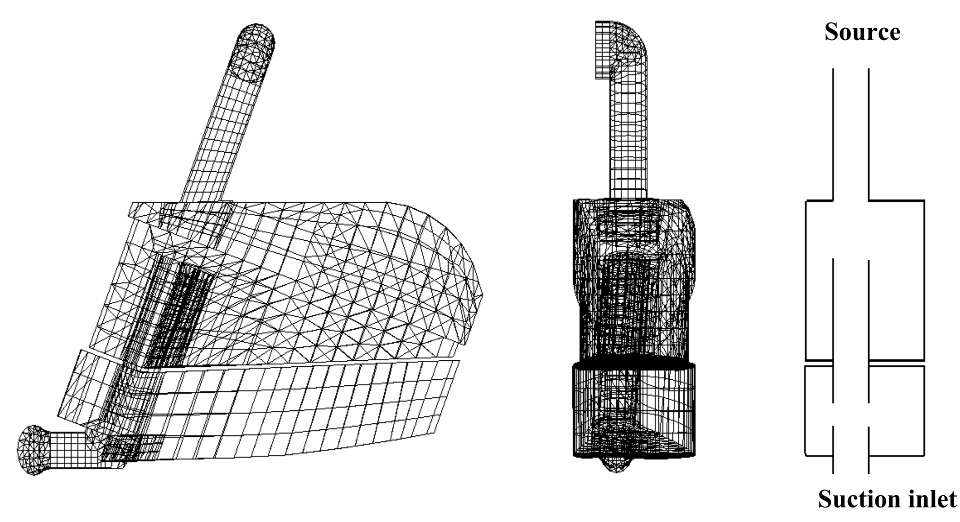

Figure 3: Boundary element and schematic model of the suction muffler.

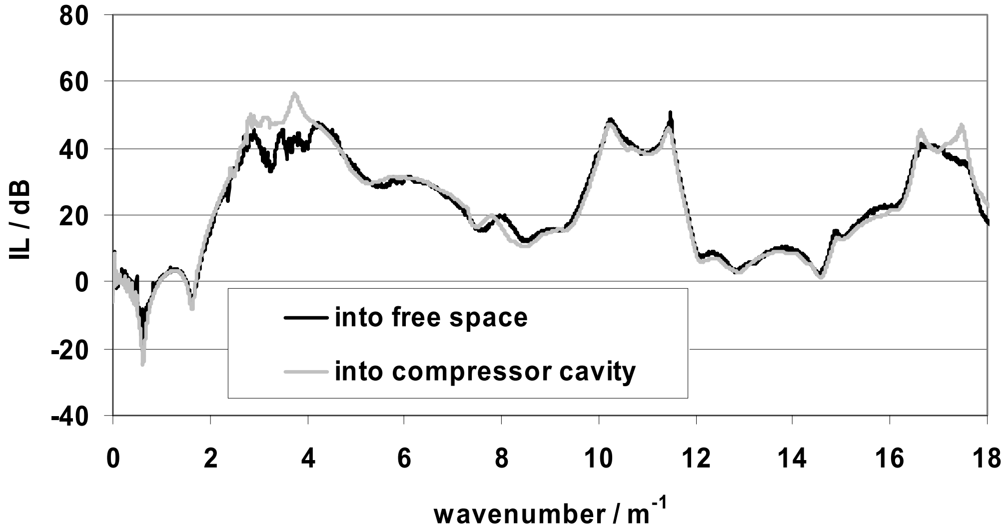

Figure 4: Measured IL in air into free space and into compressor cavity.

In a recent study (Svendsen, 2003) it has been shown numerically that the insertion loss is independent on whether the suction muffler is allowed to radiate into free space or into the compressor cavity. Generally, it can be stated, that the insertion loss is independent of the boundary conditions at the suction inlet as long as the reference source is subjected to the same boundary conditions. Figure 4 shows an experimental verification of this observation, where the muffler is radiating into a closed cavity and into an open compressor (top cap removed). It is seen, that the IL is nearly identical except from 3 to 4 m^{-1}, which is due to a low signal to noise ratio at the suction inlet of the open compressor configuration.

There are large differences in the line-shape and amplitude when comparing the numerically and experimentally obtained IL in Figure 2 and 4, respectively. This is mainly due to dissipation effects inside the muffler. This indicates that the IL is dependent on the boundary conditions inside the muffler including those of the source. Figure 5 shows a comparison between the calculated and measured insertion loss in air. In the calculation a frequency independent impedance of the muffler interior was chosen, which in this case was 60 times that of air. It is seen, that the calculated amplitude fits rather well, but the damping effects are too large to yield the observed troughs and peaks. For plastic mufflers, where the fluid-structure interaction is still weak, the impedance depends only on

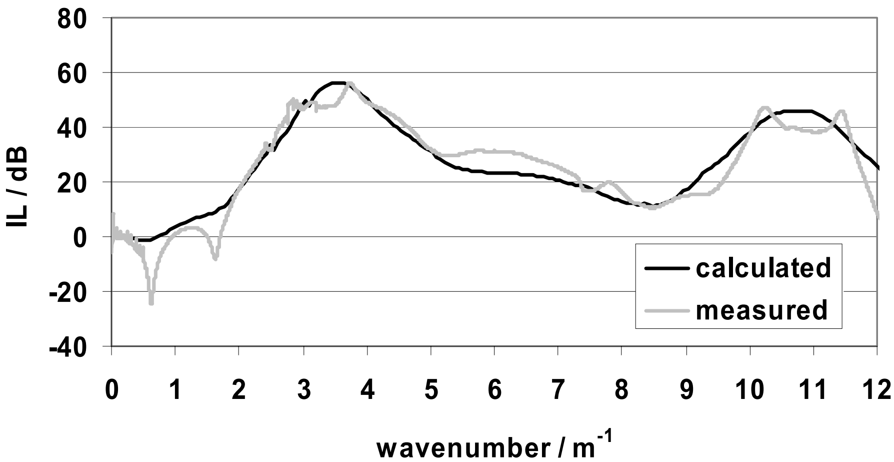

Figure 5: Comparison between calculated and measured IL.

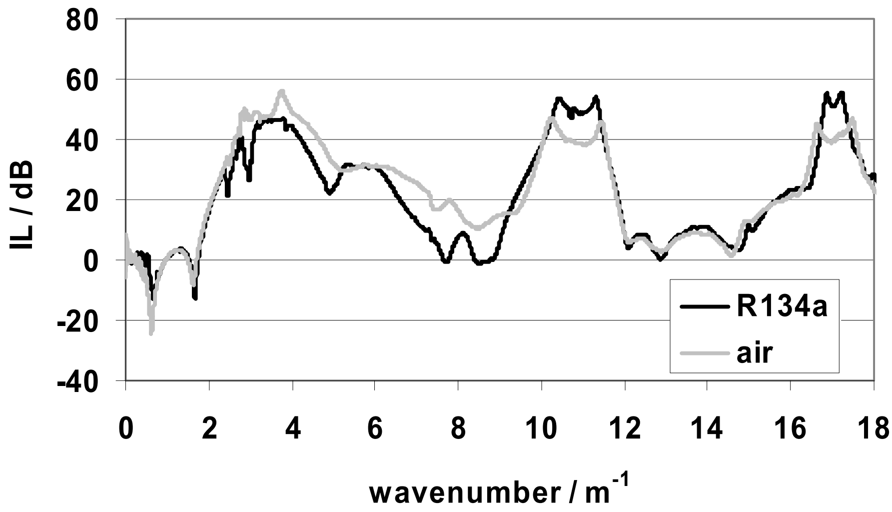

Figure 6: Comparison between measured IL in R134a and air.

structural parameters i.e. mode shapes and their damping. However, this does not mean that the IL is invariant with respect to the acoustic medium. The frequencies of the acoustic resonances inside the muffler scale with the speed of sound, whereas those of the structure are obviously independent of it. Figure 6 shows the measured IL in R134a and air. It is seen that there are some differences in the IL, which is mainly due to the fact that the speed of sound in air is around twice that of R134a. Furthermore, it is conceivable that the impedance of the source might have some significance.

5. DISCUSSION

In this study, only IL has been used for the characterization of mufflers. In the case, where the boundary conditions are well-known, IL is better suited for acoustic optimization as opposed to TL and NR. It is, however, difficult to determine both the impedance of the source and the internal impedance of the suction muffler. In principle, therefore, both TL and NR can be just as good as the IL for assessing the acoustic performance of suction mufflers. Under these circumstances good engineering practice would perhaps be to use all performance parameters simultaneously.

6. CONCLUSIONS

In summary, the following observations have been made:

- The compressor cavity is a heavily damped system, where most of the acoustic power, which is radiated into the cavity, is dissipated.
- It has been shown experimentally, that the IL is independent of the boundary conditions at the suction inlet.
- The IL is strongly dependent on the remaining boundary conditions, which includes the source impedance and the internal impedance of the muffler.

NOMENCLATURE

A	area	*S*	virtual surface
IL	insertion loss	*SW*	sound power
LDW	level difference of sound powers	*SWL*	sound power level
n	component normal to surface	*TL*	transmission loss
NR	noise reduction	*v*	particle velocity
p	sound pressure	*Z*	acoustic impedance

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