1990

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SUCTION MUFFLER DESIGN FOR NOISE REDUCTION ON HIGH EFFICIENCY RECIPROCATING COMPRESSORS

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ABSTRACT

The development of high efficiency reciprocating compressors for LBP domestic applications to E.E.R. level of 5 BTU/hW and more, requires the optimization of various components involved in compression cycle, friction losses, electric motor efficiency,etc..
In this paper is considered particularly the suction muffler design and related experimental techniques developed to optimize the noise attenuation factor and pressure losses.
The need of keeping the suction pressure drops as minimum as possible, forces in oversizing the suction ports and ducts. By this fact, some more noise is generated by the suction process.
Correct sizing of suction muffler is therefore required to reduce the suction noise emission.
An experimental technique was developed to verify many suction muffler design, without testing all the configurations on the compressor, avoiding the test repeatibility and the unavoidable instability of the noise source, i.e. the compressor itself.

REACTIVE MUFFLERS

A "silencer" could be defined as an apparatus inserted in a flow of sound wave energy to reduce the wave amplitude between two fixed section. Reactive type mufflers are generally sized as a succession of chambers of different volumes and shape connected by ducts. These types of mufflers operate by reflecting part of the energy of the wave incident on the muffler.
The study of the suction mufflers of the high efficiency reciprocating compressors, with their pipe resonators and chambers separating baffles, was carried out with simplifying assumptions of plane wave propagation in a duct with expanding and contracting section (fig 1).

The size and the section variations provide impedance mismatches for the sound wave traveling through the muffler. This fact results in a partial reflection of the sound energy back toward the source;in this way only a part of the sound energy is transmitted. The contracting section provides the same acoustic effect of the expanding section and could be calculated by the same relationship:

$$T.L.=10*\log_{10}(1 + \frac{1}{4} * (\frac{S2}{S1} - \frac{S1}{S2})^2 *Sen^2\frac{(2*PI*f*L)}{C} \quad [dB]$$

where:
f= frequency
L= length of expansion chamber
C= speed of sound of fluid
PI= 3.14
S1= smaller section; S2= larger section

SIDE RESONATOR

Some types of mufflers are composed by Helmholtz resonators connected to the side of suction passage. These resonators are formed by a cavity of volume V connected to the duct by a passage having length L and section S (fig 2). Through this passage the sound waves reach the cavity,that acts like a spring while the connecting passage acts like a mass. In other words, the resonator is equivalent to a spring-mass system with its resonating frequency given by:

$$f=(\frac{C}{2*PI}) * (\frac{S}{(L*V)})^{1/2} \quad [Hz]$$

where:
S= passage section
L= passage length
V= volume of the cavity
PI= 3.14
C= speed of sound of fluid

When at the resonator passage is present an acoustic wave with frequency coincident with the resonator frequency an high attenuation of acoustic wave energy occurs.

A computer simulation program was developed assuming plane wave sound propagation. By defining the"geometry of the muffler", the physical properties of the fluid, the frequency range, the attenuation curve is obtained.

INSERTION LOSS MEASUREMENT METHOD

Transmission Loss is one of the most widely used parameters to describe the muffler characteristics, from a theoretical point of view.
The Transmission Loss is defined as:

$$T.L.= 10 \log_{10}\left[\frac{Wi}{Wt}\right]$$

where:
Wi = sound power incident on the muffler [W]
Wt = sound power transmitted through the muffler [W]

The attenuation characteristics of the mufflers, are generally shown as curves of transmission loss vs. frequency.
From an experimental point of view it is more usual to define the attenuation characteristics of the mufflers as "Insertion Loss" curves.
The Insertion Loss is defined as the difference, in sound pressure level, measured at one point with and without the muffler inserted between the noise source and the measuring point:

$$\text{Insertion Loss} = L2-L1 \quad [dB]$$

where:
L1= sound pressure level with muffler
L2= sound pressure level without muffler

The Insertion Loss measurement involves the evaluation of two quantities:

1) static insertion loss

2) self-generated noise level

The evaluation of self-generated noise is necessary when fluid flows through the muffler, but to simplify the test procedure we measured the static insertion loss measurements, without gas flow.
The muffler under test was connected at the end of a duct, acoustically insulated. The noise source was a loudspeaker with a good frequency response in the 300- 3000 Hz. range. This source was connected at the opposite end of the duct and powered by a "white noise" generator.
All the test equipment was assembled in a semi-anechoic room (fig.3). The average sound pressure level was measured by five microphones with the noise generator working at the fixed power.The same measurement was performed without the muffler connected to the duct.
The sound pressure level was recorded by a third-octave real time analyzer and the Insertion Loss was obtained for

each third-octave band from 40 to 16000 Hz.

CHECK OF MEASUREMENT RELIABILITY

The assumed simplification to carry on the measures of the Insertion Loss without gas flow, has been checked measuring the sound pressure emitted by a compressor with and without muffler. Both air and refrigerant R-12 were used as working fluids.
In fig.4 is shown the attenuation due to the muffler measured on the compressor, in comparison with the one measured on the test apparatus; the working fluid of the compressor was air. In fig. 5 the working fluid of the compressor was R-12.
It's interesting to observe that the difference in attenuation between air and R-12 is corrected by a simple frequency shift in the ratio of the speeds of sound of the fluids:

$$F_{R12} = F_{air} * \frac{C_{air}}{C_{R12}}$$

where:
F = frequency
C = speed of sound,air or R 12

MUFFLER CONFIGURATIONS AND RELATED NOISE ATTENUATION VALUES

Some solutions have been developed with reference to schematic drawings of fig.6
The insertion loss values are reported in the tested configurations as follows:

Solution 1		Attenuation [dB(A)]
a)	-with inlet pipe and baffle	20.5
b)	-with inlet pipe -without baffle	11.3
c)	-without inlet pipe -with baffle	11.0
d)	-without inlet pipe and baffle	6.4

Solution 2

It differs from solution 1 for the substitution of inlet pipe with an elastic bellow inserted between the muffler and shell suction tube.

The attenuation values are:

a) with baffle and bellow :	18.0 dB(A)
b) with perforated baffle and bellow:	13.5 "
c) with bellow and without baffle :	11.3 "
d) without baffle and without bellow:	3.5 "

COMMENTS:

The attenuation effect of bellow in solution 2 a) is mainly in the 1/3 octave band of 400-500 and 630 Hz. The bellow reduces the compressor cavity excitation in the range of typical suction gas noise spectrum.
Similar effect is obtained by the inlet pipe (solution 1a).

Solution 3

This solution differs from solution 2 only for splitting the internal volume in two volumes V_1 and V_2 and connecting them with two openings S_1 and S_2 to the gas flow duct.
The attenuation value is 18.3 dB(A).

PRESSURE LOSSES OF VARIOUS SOLUTIONS

To compare the gas pressure losses of different solutions, the "steady" air flow, corresponding to a fixed pressure drop of 0.981 kPa, was measured. The air flow and related attenuation data are:

Solution nr.	Air flow (dm^3/sec)	Attenuation dB(A)
-1d)	$1.2833*10^{-3}$	6.4
-2d)	$1.1111*10^{-3}$	3.5
-2a)	$0.9864*10^{-3}$	18.0
-1c)	$0.9611*10^{-3}$	11.0
-1d)	$0.7917*10^{-3}$	11.3
-3)	$0.7444*10^{-3}$	18.3
-1a)	$0.7083*10^{-3}$	20.5

COMMENTS:

As expected, the noise attenuation is about inversely proportional to the "steady flow" pressure loss of the muffler. The choice among different solution is based on the compromise between these two opposite effects and the E.E.R. targets of different compressor series.

TYPICAL REACTIVE MUFFLER

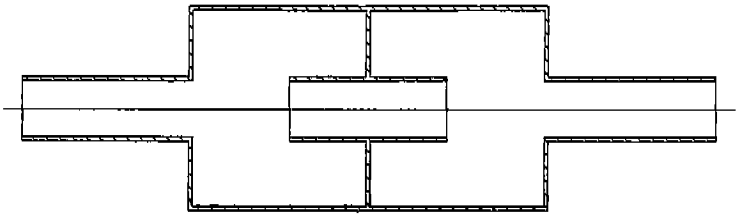


Fig. 1

TYPICAL SIDE RESONATOR

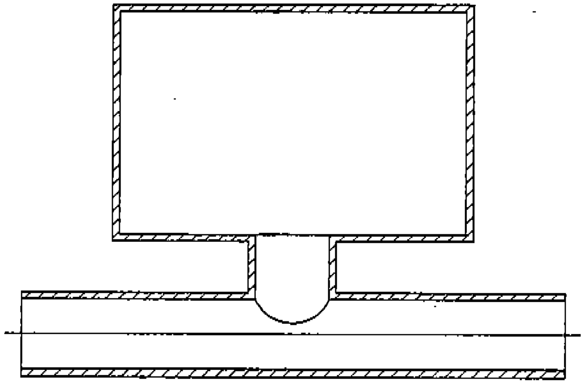


Fig. 2

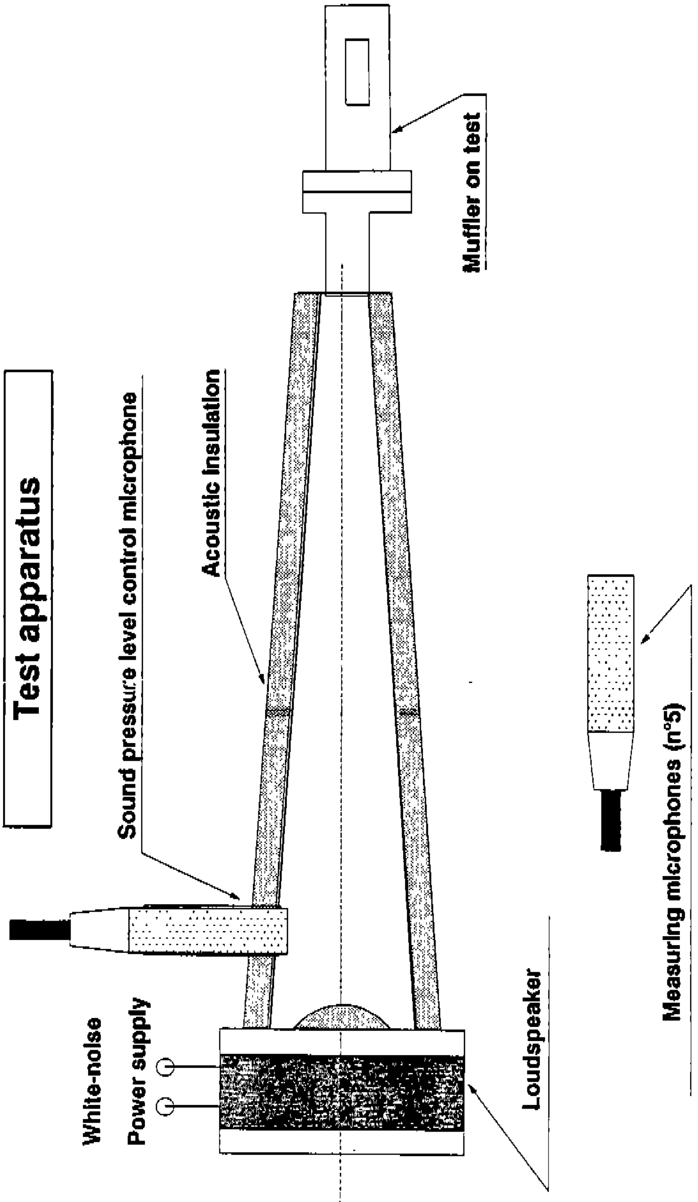

Test apparatus
Sound pressure level control microphone
Acoustic Insulation
Muffler on test
White-noise
Power supply
Loudspeaker
Measuring microphones (n°5)

Fig. 3

INSERTION LOSS CURVES WORKING FLUID AIR Solution 2 case C

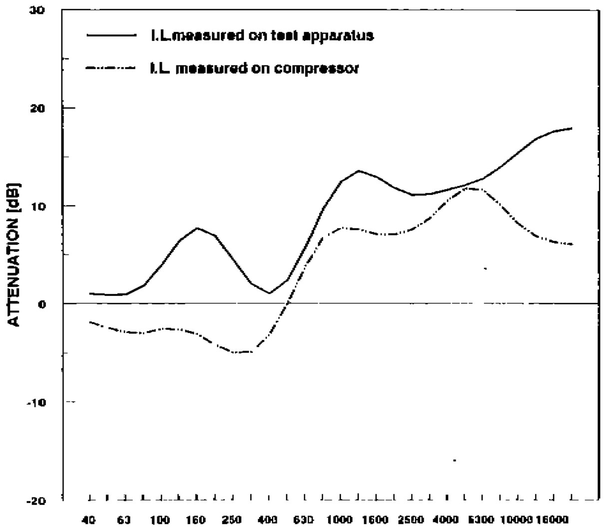


Fig.4

INSERTION LOSS CURVES COMPRESSOR WORKING FLUID:R 12 Solution 2 case C

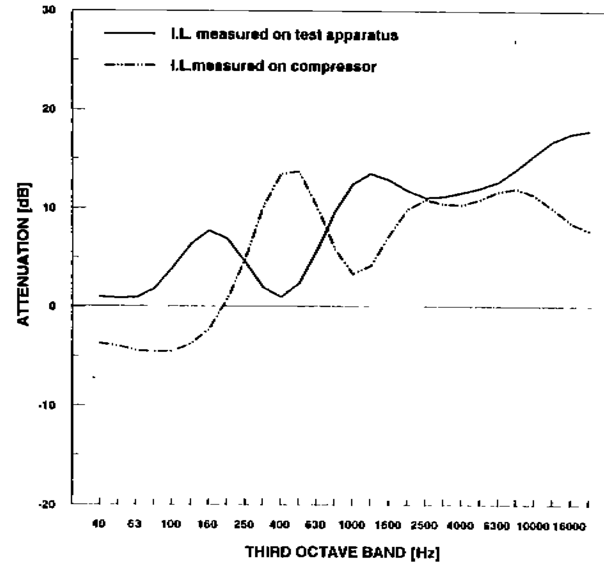
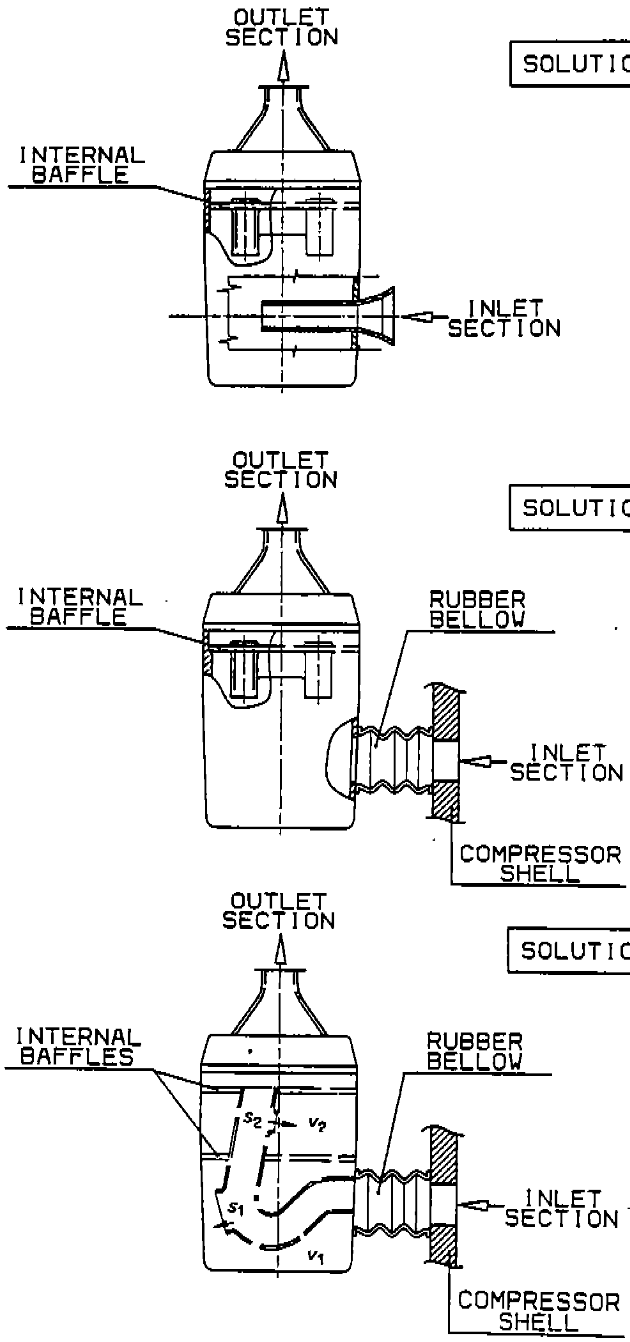


Fig. 5

Fig. 6 SUCTION MUFFLER SOLUTIONS



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