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GUIDELINES FOR EFFECTIVE USE OF REFRIGERATION DISCHARGE MUFFLERS

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1. INTRODUCTION

Discharge pressure pulsations in a refrigeration system can excite vibration of the condenser and associated pipes resulting in excessive noise. Although discharge muffling elements usually are present within the manifold or shell of a welded hermetic compressor, designers of refrigeration and air conditioning systems often include an external muffler in the discharge line to further supress pulsations. From the viewpoint of a refrigeration system designer, the most effective muffler is that which maximizes the insertion loss IL at the muffler exit over the first 10 to 20 harmonics of the compressor running speed f_0, which dominate the pulsation spectrum. The insertion loss is given by

$$IL(nf_0) = L_p(nf_0) - L^*_p(nf_0), \text{ dB}$$

where L^*_p is the sound pressure level at the muffler exit , L_p is the sound pressure level at the same point in the discharge line when no muffler is installed, and n is the harmonic index.

Different reactive muffler configurations are compared in the literature[1-4] in terms of transmission loss TL, which is the difference between the incident and transmitted acoustic intensity levels. For a given muffler configuration, however, IL depends not only on TL, but also on the acoustical impedances of the source and the load between which it is installed. This has been shown for internal combustion engines [4,5], and for reciprocating compressors [3,6,7]. Hence, the system designer has no reliable way of estimating what IL will result when a muffler of known TL is used in a particular refrigeration system without resorting to computer modeling or experiment for each new application. Such information unfortunately is not available in the literature. To fill this void, an experimental study of eleven commercially available dischsrge mufflers has been conducted. The insertion loss due to each muffler was measured in a system operated at constant source and load conditions, as was the insertion loss due to a single muffler for five different sets of thermal conditions. In addition, the effect of muffler location in the discharge line was examined. The results have been used to develop practical guidelines for designers of refrigeration systems. Some analytical issues associated with discharge muffler effectiveness are also discussed briefly.

2. COMPARISON OF ELEVEN DIFFERENT MUFFLERS

Geometric details of the eleven mufflers tested are given in Table 1. Most of these were of one of the three types shown in Figure 1: simple expansion chamber, expansion chamber with perforated baffle, or pipe resonator muffler. The mufflers were tested in a R-22 system driven by a two-cylinder welded hermetic compressor with f_o= 57.5 Hz. The discharge line between the compressor and the condenser was 19.05 mm in diameter, and 3835 mm long. Each muffler was installed with its inlet 1270 mm away from the compressor discharge port, and pressure pulsations were measured 431.8 mm downstream from the muffler inlet. Both for the reference case when no muffler was installed and for each of the muffler evaluations, the system was operated under thermal conditions given by case III in Table 2, so that the source and load impedances were the same for each case.

Results of these tests are shown in Figures 2-6. These Figures represent line spectra of IL for n=1 to 10, but the points at each harmonic have been connected by straight lines to help distinguish results for one case from those for another. It is remarkable how similar the IL spectra for different mufflers are. The results show that all of the mufflers produced an IL of roughly 20 dB for n=2 to 5, with those of greater area expansion ratios achieving slightly higher performance. In fact, Figures 2-6 show that all mufflers of similar area expansion ratio produced nearly identical IL, and that the most signifigant performance differences among mufflers generally correspond to differences in expansion ratio rather than differences in internal geometry. All except muffler K yielded very little attenuation at the seventh and eighth harmonics, and and all but K and C produced IL of 30 to 40 dB at the ninth harmonic. Muffler K was unique among those tested as it had a feature which permitted the areas of openings in several internal baffles to be varried. Two tests were performed for this muffler in either extreme of adjustment and are labeled K_1 and K_2. As Figure 6 reveals, this feature resulted in negligible changes in IL for all but the ninth harmonic. Additionally, all of the mufflers caused amplification of discharge pulsations at the first harmonic. This is consistent with the observations of other investigators [3,6,8].

In Figure 7 the insertion loss due to muffler A, a simple expansion chamber, is compared to its transmission loss predicted by theory [1]. Note that TL corresponds roughly to the mean value for IL over the frequency range considered, but differences as great as 20 dB exist . Hence, TL may not provide the designer with a good estimate of the IL expected when a particular muffler is installed in a specific refrigeration system.

3. OTHER FACTORS INFLUENCING MUFFLER PERFORMANCE

In addition to variations in muffler geometry, the effects of the system thermal operating conditions and the location where the muffler was installed were examined. The insertion loss due to muffler C for each of the five sets of thermal conditions given in Table 2 are shown in Figure 8. Observe that the IL spectrum for each of the five thremal conditions is distinct. At some harmonics, differences between IL spectra as great as 45 dB are seen. Also note that each harmonic of the IL spectrum is affected differently by changes in thermal conditions. These differences are much greater than those seen in Figures 2-6 for different mufflers tested at a single thermal condition.

The insertion loss due to muffler A was measured when the system was operated at thermal condition III and the muffler was installed at two different locations in the discharge line. The muffler was tested while installed with its inlet 1270 mm and 1448 mm away from the compressor discharge port. Results from these tests are reported in Figure 9. The change in location produced signifigant changes in the IL spectrum for n=1 to 3 and for n=5 to 6, with differences in some cases as great as 15 dB. While some differences in IL are observed at the higher frequencies, they are not as signifigant.

4. CONCLUSIONS AND DESIGN GUIDELINES

The eleven discharge mufflers tested, with expansion ratios of 16 to 58, were found to have roughly equal performance in attenuating discharge pulsations. The insertion loss was found to increase slightly with increases in area expansion ratio, and the effects of changes in internal details were virtually negligible. This suggests that a system designer would be wise to use the simplest configuration of all: the simple expansion chamber. The results indicate that an area expansion ratio of 20 to 30 and a length of 100 to 150 mm should produce IL of nominally 20 to 30 dB for frequencies between 100 and 600 Hz. More complicated configurations do not seem to provide much additional benefit, although theory predicts that they should have greater TL, at least over a narrow band of frequencies [1].

In contrast, changes in thermal conditions and muffler location produced marked changes in muffler performance. Farstad and Singh [8] have developed a method for modeling the effect of thermal conditions on the acoustical load impedance of the condenser, and Farstad [9] has shown that the impedance presented by the condenser to the muffler (or compressor) is strongly dependent on thermal operating conditions. This is probably the reason for the large differences among the IL spectra of Figure 8. Further work is needed to better quantify this issue.

Muffler effectiveness was also found to be sensitive to the location in the discharge line where the muffler is installed. This might suggest that there is an optimum location which would maximize muffler performance. If a refrigeration system were always to operate under the same thermal conditions, this might be the case. For a system subjected to variable thermal conditions, however, the location in the condenser where the refrigerant condenses fully is variable. In fact, this effect is largely responsible for the change in condenser acoustical impedance with thermal conditions [9].When thermal conditions are changed, it is as though the distance between the condenser and the muffler has been changed. Consequently, an optimum muffler location for all thermal conditions may not exist. This is in contrast to the use of mufflers in engines or air compressors, where the acoustical impedance of the load presented to the muffler remains nearly constant. This result, unique to refrigeration systems, should be helpful to system designers.

Aknowledgement

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Table 1. Geometric Details of Mufflers Evaluated

Muffler Designation	Area Expansion Ratio, S_2/S_1	Length, L	Reference Figure	Internal Details
A	23.8	98.6 mm	1(a)	------
B	23.8	98.6 mm	1(b)	------
C	16.0	127.0 mm	-------	Three baffles each with single hole.
D	33.1	88.9 mm	1(a)	------
E	31.8	114.3 mm	1(a)	------
F	31.8	114.3 mm	1(b)	------
G	31.8	114.3 mm	1(c)	L_i=22.9 mm d_i=12.7 mm
H	33.1	111.3 mm	1(a)	------
I	33.1	111.3 mm	1(c)	L_i=31.8 mm d_i=12.7 mm
J	33.1	111.3 mm	1(c)	L_i=22.2 mm d_i=9.5 mm
K	58.1	171.5mm	-------	*

* Four baffles with adjustible area openings, three 6.3 mm dia. tubes connecting volumes at ends.

Table 2. System Thermal Operating Conditions

Condition Designation	Refrigerant Flow Rate (kg/hr)	Suction Pressure (kPa gage)	Discharge Pressure (kPa gage)	Condenser Exit Temp. (K)	Compressor Inlet Temp. (K)
I	17.2	68.9	1723	312	311
II	18.1	103.4	2481	330	311
III	97.1	344.7	1723	316	289
IV	108.8	413.6	2619	333	289
V	111.6	344.7	1206	298	289

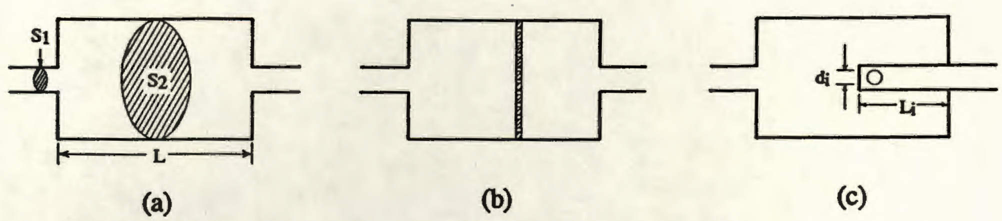

Figure 1. Three types of mufflers: (a) Simple expansion chamber (b) Expansion chamber with perforated baffle (c) Pipe resonator muffler.

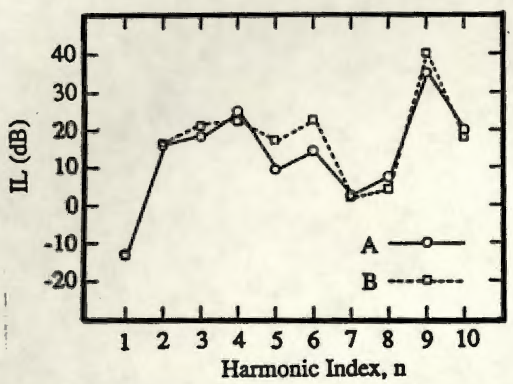

Figure 2. Insertion loss due to mufflers A and B.

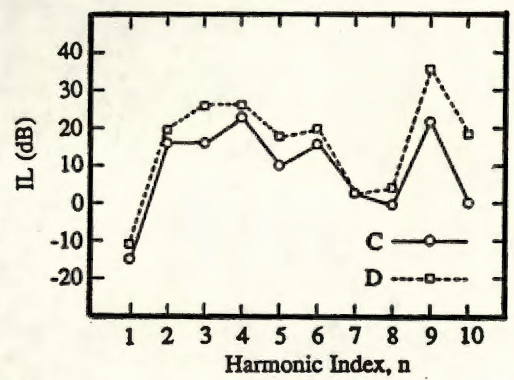

Figure 3. Insertion loss due to mufflers C and D.

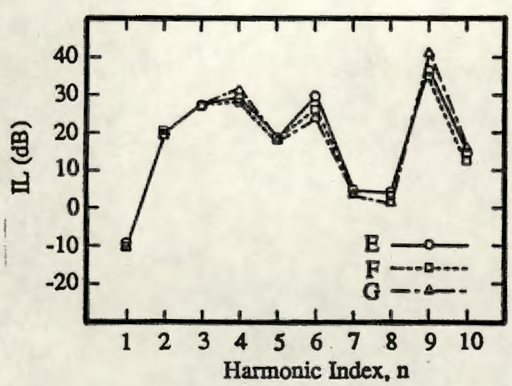

Figure 4. Insertion loss due to mufflers E, F, and G.

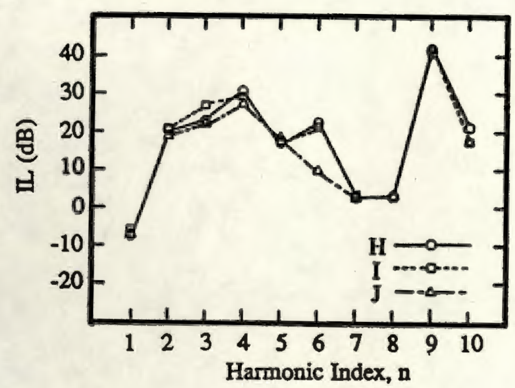

Figure 5. Insertion loss due to mufflers H, I, and J.

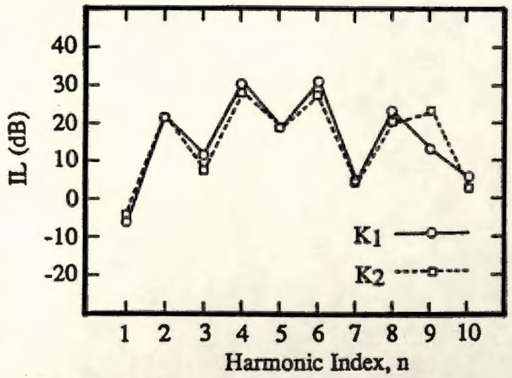

Figure 6. Insertion loss due to muffler K.

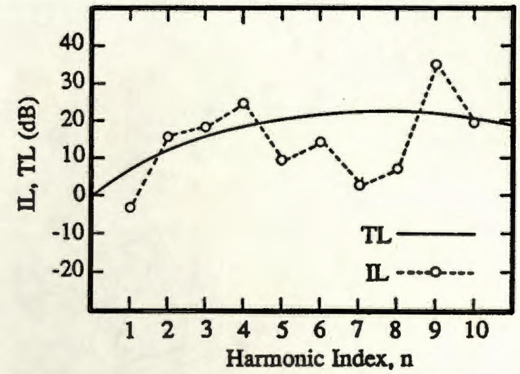

Figure 7. Comparison of IL and TL for muffler A.

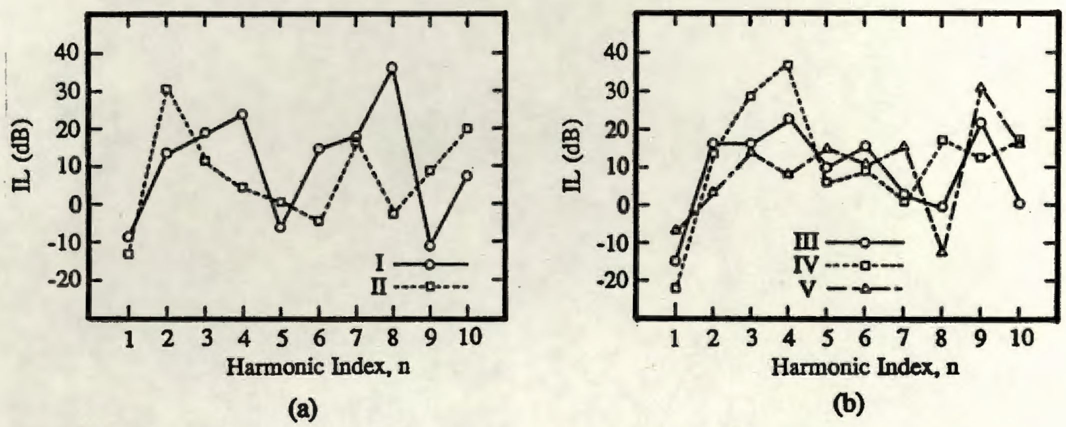

Figure 8. Insertion loss due to muffler C for the five sets of thermal conditions given in Table 2. (a) Conditions I and II. (b) Conditions III, IV, and V.

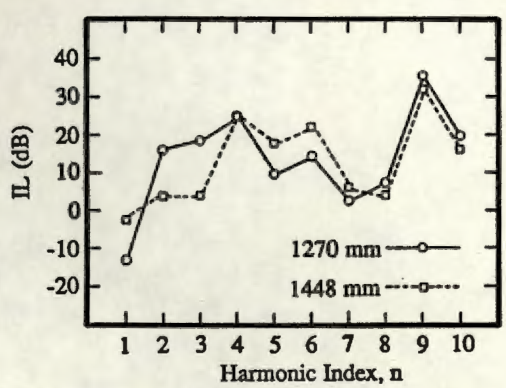

Figure 9. Insertion loss due to muffler A for two different installation locations