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Analysis of Noise Emission of A 35CFM Capacity Air Compressor

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35CFM CAPACITY AIR COMPRESSOR

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1) INTRODUCTION

Compressor manufacturers are continually confronted with the requirement of reduced noise emission of their compressors.

Most approaches are confined to secondary measures, i.e. acoustical enclosures which by themselves can cause a series of problems (for example high temperature). For these reasons it is desirable to achieve a noise reduction through design changes at the noise source.

Our objective on this project was to determine which parts of the air compressor machinery contribute substantial portions of the total sound pressure, and whether the noise emission can be changed by modification of the compressor valves.

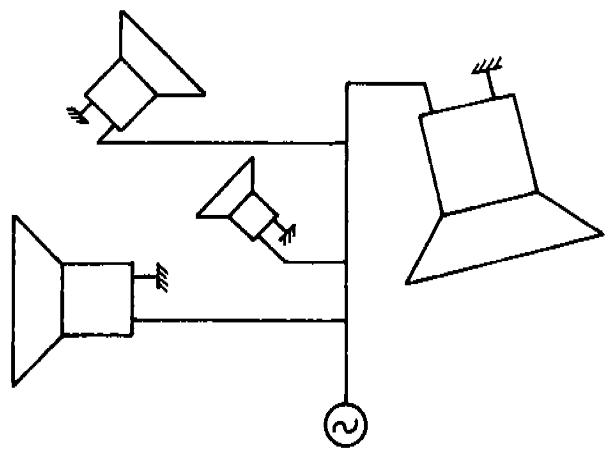
2) TEST APPARATUS AND EQUIPMENT

The test compressor was a single cylinder air compressor with a bore of 134 mm and a stroke of 85 mm. A rotative speed of 950 RPM and a compression ratio of 7 were selected. The compressor was mounted in a free field room on an inert base. The first natural frequency of the system (compressor/base) was 4.5Hz. The DC-motor driving the compressor through an axle with constant velocity joints and elastic couplings was mounted outside the test room.

The test room had a volume of approximately 200 m^3 and free-field conditions above a frequency of 125Hz. For noise measurements 1" free-field microphones were used positioned at a distance of 1 m from the compressor surface and 1.5 m above the floor. The mean sound pressure level was established from eight individual measurements of each test setup.

3) APPROACH

In our opinion the compressor can be substituted by an arrangement of loudspeakers each one emitting sound of different intensity and frequency spectrum.



It is known that the noise level originating from two sources of different intensity is determined by the source of higher intensity.

This fact establishes the approach used to recognize the noise sources which are the major contributors to the total noise levels in the above loudspeaker model. By acoustical shielding of single elements and simultaneous observation of the total noise level, it is possible to establish the important individual noise sources.

To establish priorities of the individual noise sources, all elements in the system have to be acoustically shielded. By removing the shielding from individual loudspeakers and registering the effect on the basic noise level, priorities can be established.

The above theories were the basis of our investigations.

4) TEST PROCEDURE

4.1) Original Compressor

First, the noise emission of the uninsulated compressor was measured. Standard intake filters were used and the air intake was inside the test chamber. The compressed air was delivered into a receiver outside the test chamber.

4.2) Compressor Completely Shielded

In accordance with the model of shielded individual loadspeakers, the whole compressor including the foundation was covered with a maximum "sound package". Mats which consisted of an absorption layer of 20 mm and a septum with a weight per unit area of 5 Kp/m^2 were used.

The intake air was taken from the outside through flexible piping.

Diagram D1 shows the average sound spectrum of the original and the shielded compressor.

The noise level of the two measurements differs by 24dB(A).

During the tests it was impossible to acoustically insulate the individual noise sources as suggested in the theoretical model. Consequently, direct conclusions as to the importance of the individual sources are not possible. The approach taken was a step by step modification of the "sound package".

4.3) Cylinder Head Uncovered

By removing the insulation from the cylinder head, the total noise level rose by 4.5dB(A).

4.4) Cylinder Uncovered

By further removal of the insulation from the complete cylinder, the noise level increased by another 1.5dB(A).

4.5) Crankcase And Fan Uncovered

Freeing the crankcase and fan from insulation increased again the noise level by 12dB(A).

4.6) Replacement Of Fan By A Simple Disc

The fan casing was removed and the impeller, also functioning as a flywheel, was replaced by a simple disc. This version reduced the noise level by 7dB(A) as compared to the previous version.

4.7) Fan Casing Removed

This modification, with unshielded crankcase and original impeller, but without fan casing, resulted in a noise level 3dB(A) higher than the one above.

4.8) Foundation Uncovered

Removing the insulation from the foundation and reinstalling the original fan casing, resulted in no change in noise level compared to the version tested in 4.5.

4.9) Discharge Piping Uncovered

No change was measurable with this modification.

Following is a listing of the absolute measured values of the average sound pressure in the steps discussed above:

#	Conditions	Sound Pressure dB(A)
2	Total sound insulation	60.5
3	Free cylinder head	65.0
4	Free cylinder	66.5
7	Free crankcase, impeller replaced	71.5
6	Fan casing removed	74.5
5	Free crankcase, original fan	78.5
8	Free foundation	78.5
9	Free discharge line	78.5
1	Original compressor, no shielding, intake in test room	84.5

5) DISCUSSION OF RESULTS

The results show the major noise contributor to be the intake noise.

The second most important noise source is the fan. The fan casing especially has an unfavorable effect. The sound pressure caused by the casing is higher than the one from the impeller together with all other compressor components.

The third most important contributor seems to be the crankcase. This conclusion is valid since the contribution of the foundation or the discharge line has to be at least 8dB(A) lower than the value where their addition resulted in no change of total sound pressure. Discharge line, as well as the foundation, can therefore have a maximum level of 70dB(A).

A more detailed study of the single loudspeaker "discharge line" in its immediate surrounding resulted in a sound pressure reduction of 23dB(A). Graph D2 shows a comparison of the mean sound spectrums of the shielded and unshielded discharge line.

The total noise contribution of the cylinder and cylinder head on the test compressor has to be considered as negligible.

Based on the above it becomes obvious that the effect of valve modifications, even after elimination of intake and fan noise, cannot be recognized by sound pressure measurements.

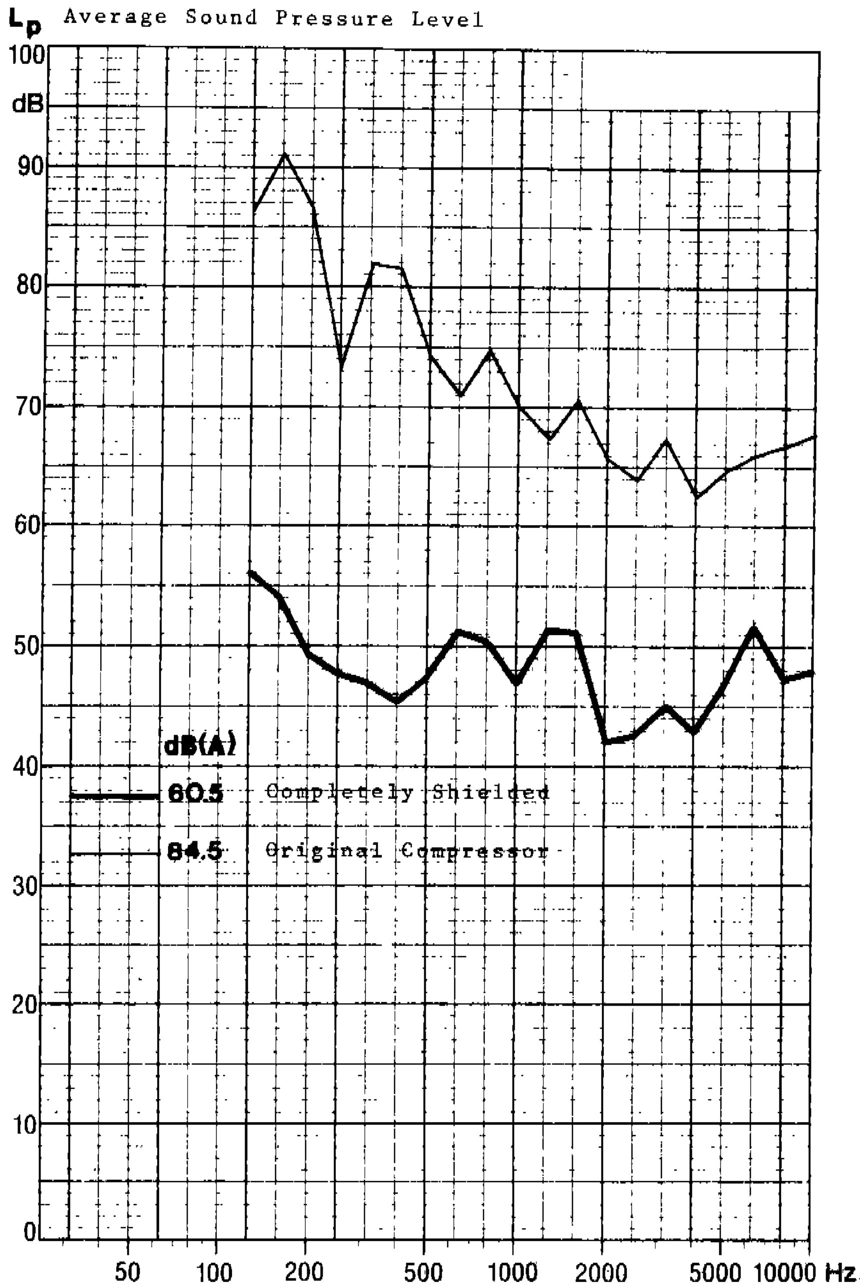


Diagram D1: Comparison of frequency spectra of the original and completely shielded compressor.

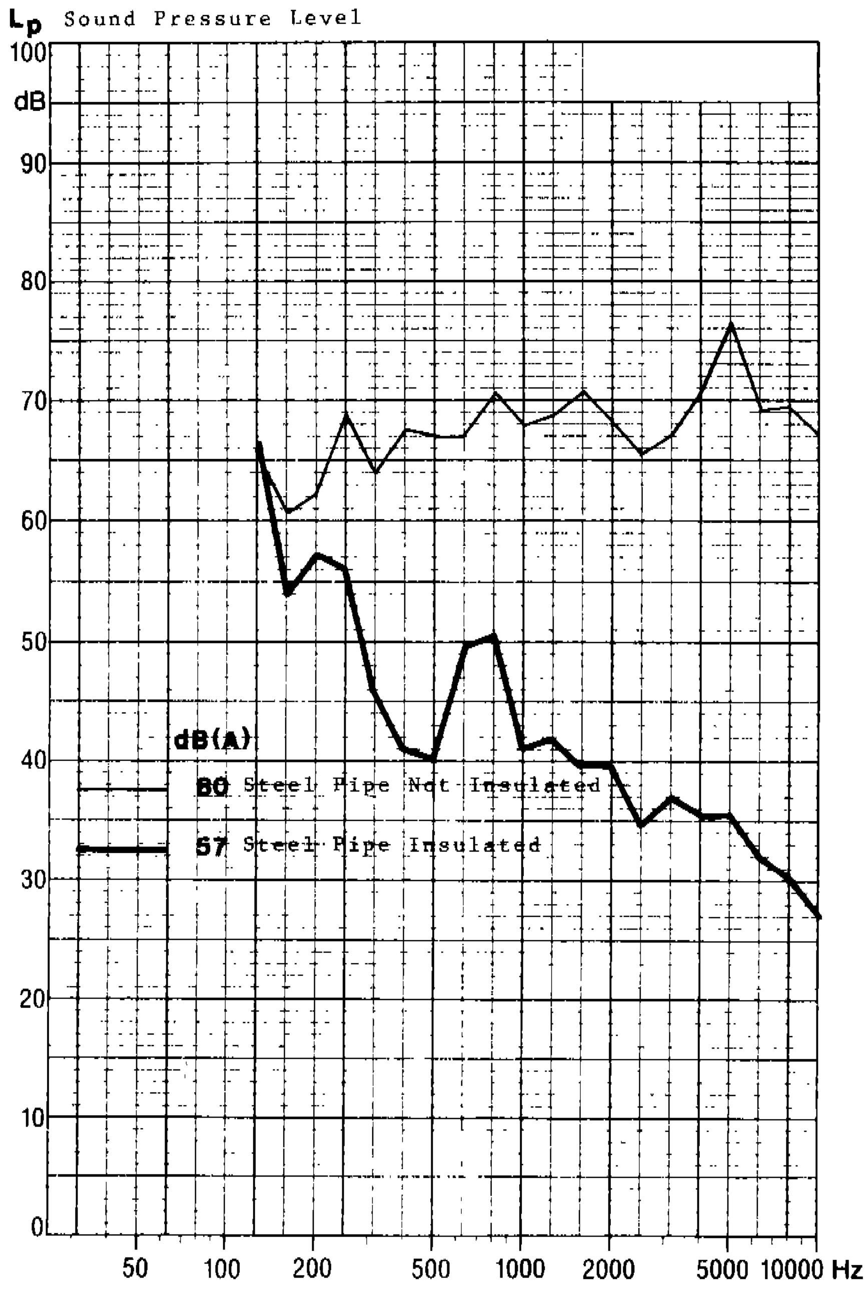


Diagram D2: Comparison of frequency spectra in close proximity of the discharge pipe in non-insulated and insulated condition.