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Higher Efficiency, Lower Sound, and Lower Cost Air Conditioning Compressors: Part 2 - Sound/Discharge Pulse

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ABSTRACT

This two-part paper describes a compressor redesign program for improving efficiency, lowering sound emission, and cost. Part 2 indicates how sound and discharge pulse were reduced through research conducted on transmission loss of plastic vs. steel manifolds and discharge pulse modeling using four pole parameters.

Common reciprocating compressors utilize steel flapper valves and a Cylinder Head that encloses a suction and discharge plenum. In these compressors, the suction gas enters the cylinder head into the suction plenum, passes through a steel flapper valve into the cylinder on the suction stroke, is compressed on the discharge stroke and is forced through a steel flapper valve into the discharge plenum. Heat transfer occurs from the high temperature discharge plenum to the low temperature suction plenum across the cylinder head wall which separates the plenums.

In the Inertia compressor the gas flow and valve system is improved, resulting in cooler gas entering the cylinders and reduced flow losses through the valving. The suction gas is routed through the crankcase wall, through the piston, and enters the cylinder past a polymeric suction valve that floats freely on top of the piston. The free-floating valve results in increased valve open time and decreased flow losses. On the discharge stroke the gas is compressed and forced past a polymeric discharge valve that allows a large flow area with significantly reduced re-expansion volume. The cylinder head on this compressor is devoted entirely to the discharge plenum. The result of the cooler suction gas into the cylinder and reduced flow loss through the valve system is a very high-efficiency reciprocating compressor. The trade off for the efficiency improvement is in higher air-borne sound and higher cost than with Bristol Compressors standard product line, although the Inertia compressor remains very competitive in both sound and cost with competitive products.
The challenge presented was to design a compressor with efficiency near equal to the Inertia but with lower sound and cost. In general, efficiency gains normally come at the expense of sound and cost. In approaching the challenge the initial steps involved maximizing efficiency within the marketing constraints of compressor size, the objective being to maintain the overall size and mounting characteristics as the standard steel flapper valve product currently in production. The compressor size range involved with this development is 1-1/2 to 3 Ton capacity. After efficiency targets were achieved, sound characteristics and discharge pulse characteristics were evaluated and programs were implemented to bring air-borne sound levels and discharge pulse in line with objectives while maintaining efficiency improvements. The costs of various alternatives were evaluated throughout the process.

Research and development to achieve this objective has been completed and the resulting design is patent pending. Compressors have undergone extensive reliability qualification testing and the design was released to production in November 1997. This paper will summarize the design process which included extensive development and testing at Bristol Compressors and research conducted on behalf of Bristol Compressors by consultants [1] [5]. Also, additional related references [2], [3] are listed for the interested reader.

EFFICIENCY IMPROVEMENT

Initial design consideration was based on an analysis of the flow path within the compressor. In addition to the changes in flow area throughout the compressor, the suction and discharge plenums in the cylinder head were separated by use of a plastic injection molded suction manifold.

Confirmation testing of the design changes confirmed the efficiency improvements expected. A summary of the efficiency improvements can be seen in Table 1. The efficiency is stated in terms of EER (Energy Efficiency Ratio, BTU/Watt Hr).

<table>
<thead>
<tr>
<th>33000 BTU Model</th>
<th>Condition (Evaporator Temp./Condenser Temp.)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>45°/130°F</td>
</tr>
<tr>
<td>Std. Production Compressor</td>
<td>10.3</td>
</tr>
<tr>
<td>Improved Compressor</td>
<td>10.9</td>
</tr>
<tr>
<td>Percent Improvement</td>
<td>5.8%</td>
</tr>
</tbody>
</table>

TABLE 1
While efficiency improvement objectives had been met with a plastic injection molded suction manifold and a standard production discharge muffler, both air-borne sound power levels and discharge pulse were significantly increased. Measured sound power levels were 82.5 dBA (8.5 bels) and discharge pulse levels were 5.04 psi peak-peak (99.0 dBL, 9.1 bels). These problems were resolved separately as follows:

**SOUND POWER**

In an attempt to resolve the sound power problem with the plastic suction manifold, a variety of helmholtz resonator schemes were modeled and tested without complete success. Dr. Richard Lyon, RH Lyon Corp. was consulted for further analysis of the manifold [5]. The objective of the study was to determine the effects of damping and mass loading on the suction manifold. Both plastic and steel manifolds were investigated. Figures 1 and 2 show the plastic and steel manifold assemblies respectively. Changes in transmission loss and loss factors were measured in various configurations in order to determine the effects of adding mass, increased damping, and increased stiffening. By measuring sound levels inside and outside a reverberant room filled with white noise, comparisons of relative transmission losses for various configurations of the manifold were possible.

Transmission loss measurements were made by applying reciprocity. Rather than inserting a known sound source into the manifold, the sound source was placed outside the manifold in a reverberant room. Microphone measurements inside the manifold were taken (0 to 3.2 kHz) and subtracted from measurements taken outside the manifold. The result is directly proportional to transmission loss with the exception of an unknown offset. However, when comparing losses to the baseline, the unknown constant is eliminated. Fig 3 shows a comparison between the steel and plastic manifold with the transmission loss for the steel muffler (.028" thick) about 20 dB higher than for the plastic part.

For damping experiments, metal tape was added to the large surfaces of the plastic manifold. In order to insure that damping was increased, vibration decay times were measured before and after the tape was added. It was found that the tape doubled the decay times and that a transmission loss increase of not more than 6 dB could be expected.
Stiffening of the plastic was achieved by adhering nylon ribs to the outside surface of the plastic and mass loading was achieved by adding lead particles to the surface of the manifold. Figure 4 shows changes in transmission loss for the plastic manifold. Figures 4 and 6 show a small overall increase in transmission loss for increased damping and stiffening respectively. However, Fig. 5 shows that transmission loss can be dramatically increased by mass loading the part. It was found that the effect of mass loading can be increased by increasing the mass. However, as can be seen in Fig. 7, when the mass loaded plastic is compared to the steel manifold, the steel manifold still has better transmission loss.

The results of this study indicated that the plastic manifold mass would need to be increased by a factor of 5 times to be equal to the .028” steel manifold and that significantly less gains could be expected from adding damping or stiffness. Based on this study a .062” thick stamped steel manifold was produced and tested with results confirming the findings above. Fig. 8 below shows the results of the new steel manifold design as compared to the plastic manifold. The steel manifold design is cost competitive with the original plastic design and much more cost effective than an increased mass plastic part as well as being more robust.
As stated earlier, the discharge pulse measured with initial models using a production muffler (BQ Muffler) was 5.04 psi peak-peak (99.0 dBL, 9.1 bels). The objective for this design was a discharge pulse below 99 dBL and 8.5 bels. It is important to reduce discharge pulsations in the discharge plenum and exit tube because these pulsations will excite structural vibrations that travel to the condenser and result in unacceptable air-borne sound. In order to model the system a general 3-D four pole parameter concept developed by Lai and Soedel [1] was extended to a 2-D analysis to enable more efficient computation and solution of geometries which are difficult to solve in 3-D. While the details of the analysis are beyond the scope of this paper, the basic methodology for the analysis is as follows [6].

Acoustic elements, in this case the cylinder head discharge plenum, the discharge muffler, and the exit tube, are assumed attached to other acoustic elements by simple matrix manipulation. Four pole parameters, widely used for the investigation of composite acoustic systems with multiple elements were utilized. For this project the basic equation used for the derivation of four poles is a linear wave equation, assumed valid if the pressure pulsation is below approximately 15-18% of the mean pressure, and a monopole point source, valid provided the input/output port dimension is small compared to the dimension of the acoustic element. The analysis was performed using MATLAB software on several muffler models and compared to a current production muffler (labeled BQ Muffler) for which extensive test data was available. The muffler chosen for further testing and optimization is labeled 086MAX Muffler. The results for these two mufflers can be seen below in Figure 9 below. As can be seen from the model, it was expected that the 086MAX Muffler would result in significantly lower discharge pulse, particularly in the 1000-1800 Hz range.

Extensive confirmation testing was then performed using the 086MAX Muffler design with typical results shown in Figure 10 as compared to the baseline BQ Muffler. As can be seen from Figure 10, there is significant reduction in discharge pulse in the 1000-2000 Hz range as predicted from the model. The design objective for discharge pulse was met with this muffler with no significant loss in efficiency. A Taguchi design optimization study was then performed based on the 086MAX Muffler design to further refine the design for optimum efficiency.
CONCLUSIONS

- Efficiency targets were achieved through improved gas management, reduced flow restrictions, and improved valve design.

- Sound targets were achieved through redesign of the Suction Manifold, based on Transmission Loss analysis, from a plastic injection molded part to a stamped steel part, and through optimization of the suction attenuation device. It was shown that a reduction of over 6 dBA was achieved using steel (.062" thick) vs. plastic (.080" thick) manifolds (FIG. 6).

- Discharge pulse reduction with no significant loss in efficiency was achieved through redesign of the Discharge Muffler based on four pole parameter modeling (FIG. 7 and FIG. 8).

REFERENCES


