Design Modifications for Use of Hermetic Compressors in Mobile Applications

V. S. Anantaparitula  
*Siel Refrigeration Industries*

V. K. Rao  
*Siel Refrigeration Industries*

S. A. Sundaresan  
*Siel Refrigeration Industries*

K. Venkateswarlu  
*Siel Refrigeration Industries*

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Design modifications for use of hermetic compressors in mobile applications

Dr. V. Surya Anantapantula, V. Koteswara Rao, S.A. Sundaresan and Dr. K. Venkateswarlu

Siel Refrigeration Industries
Hyderabad-500 037, India

ABSTRACT

Hermetic compressors used on mobile units, such as military vehicles, submarines or air-conditioned railway coaches, have to be designed to meet extraordinary vibration, shock and environmental loads. This paper presents the additional requirements in such applications and design changes that were successfully applied to meeting such challenges on a 1.5 ton compressor for air conditioning of electronic equipment in all terrain vehicles.

Included in the development process is the use of finite element analysis to predict static stresses and cycles to failure due to impact loading for an internal mounting bracket. The final design is easy to manufacture and the additional work to productionise it was kept down to a minimum. A major advantage of the new design is that there is no reduction of refrigeration capacity or EER.

INTRODUCTION

Hermetic compressors used for air conditioning or refrigeration purposes on mobile units, such as radar-equipped or other military vehicles, submarines or air-conditioned railway coaches, have to be designed to meet extraordinary vibration, shock and environmental loads. This paper presents the additional stringent needs for air conditioning of electronic equipment in all terrain vehicles and changes implemented on a 1.5 ton hermetic reciprocating compressor designed to meet those needs.

The design that worked well for stationary air conditioning applications failed on these mobile units because the hermetic compressors could not withstand the shocks and vibrations as the vehicle moved through rough terrain. Common failures reported were failure of the discharge loop inside the hermetic shell, failure of internal suspension springs and mounting pads (compressor feet) and/or failure of system tubing resulting in a loss of charge. The design had to account for such failures and pass stringent laboratory tests in addition to extensive road tests.

Laboratory tests which simulate rough terrain include vibration/shake table, bump table and environmental tests. On the vibration table, the compressor must test normal after loading for 2 hours at any frequency in the range of 5 - 30 cycles/second with a constant amplitude of 1 mm. On the bump table, the compressor should withstand 4000 bumps at the rate of 2 bumps/second, each bump being a "free fall" of 15 to 25 mm. Environmental requirements were that the compressor had to function normally 2 hours after a soak in an ambient of 70°C, or -20°C or 40°C with 95% R.H. as well as withstand...
desert conditions. Road tests comprised of alternate cycles of 100 km of plain roads followed by 50 km of cross-country/desert roads.

DESIGN MODIFICATIONS
The mounting method inside the shell was changed from a spring supported suspension to a rigid mount (brackets welded to the lower housing) using a special arrangement to facilitate assembly. Three internal mounting brackets shoulder the load of the entire internal assembly equally. The next section outlines the use of finite element modeling to estimate static and failure stresses to decide on a suitable design. This change accounted for discharge loop and suspension spring failures.

The development of more enduring feet was another design modification. The cantilever length between the weld joint to the lower shell and the weight support was reduced. The shape and length of the four feet were then modified to enforce this concept.

Vibration transmission to the base was also taken into account and vibration absorbing pads were considered. But, the kit manufacturer had decided to use such pads for the entire appliance, and as a result, this was unnecessary.

The final design was easy to manufacture and no major changes were required in assembly. A major advantage of the new design is that there is no reduction of refrigeration capacity or EER (energy efficiency ratio).

MODELING OF INTERNAL MOUNTING BRACKET
The brackets that shoulder the weight of the internal assembly are bolted to the assembly and welded to the lower housing of the compressor. The brackets are made of a cold rolled low carbon steel sheet (0.126" thick) with a tensile strength of 44.43 ksi.

Figure 1 shows a sketch of two designs that were analyzed using finite element modeling. Part (b) of the figure shows the first design considered. Part (a) shows a change in the width of the bracket (at the bend) from 0.25" to 0.5". In both cases, the bracket is rigidly clamped in the rectangular area ABCD shown in the figure. Weight of the internal assembly (estimated at 15.43 lb per bracket) is felt on the circumference of the hole and constituted the only loading condition on the bracket.

Due to the symmetry about the vertical axis, only the left half of the bracket was modeled in the finite element package to determine stresses at various locations. 8-noded second order quadrilateral elements of the 3-d general shell type were used. This type of element can account for bending as well as transverse shear deformation and is well suited to modeling moderately thick to thin shell structures. A 2-d geometry of the plan view was used. The vertical dimension of the bracket at the area WXYZ (shown in the figure) was accounted for as thickness for the elements in that region. All other elements had sheet thickness. This allowed for a simple but effective model of the actual bracket. A graded mesh (grading factor of 1.9) was used to get a higher density mesh near the hole and at the bend while keeping the total number of nodes and elements to a minimum. The finite
element mesh had 363 nodes and 102 elements for the 0.25" design and 338 nodes and 93 elements for the 0.5" design. In both cases, a variety of standard element checks (e.g. distortion index) were made to ensure good results from the analyses to follow. For material properties, elasticity modulus of 29,000 ksi and Poisson's ratio of 0.303 were used.

ANALYSIS AND BUMP TEST RESULTS

Deformation and stress results of the static analyses carried out at (i) static load (ii) bump test load (impact loading estimated at "3g") as well as results of the fatigue failure analysis due to alternate cycles of loads in (i) and (ii) are shown in Table 2. Deformation of the bracket due to static load, plotted in Figure 2, clearly shows expected behavior. Von Mises stress contours at the 3g impact load have been shown in Figures 3 and 4 for the two design cases. The Von Mises stress of 17.023 ksi and max principal stress of 18.626 ksi as compared to the typical fatigue limit of 0.4 times the tensile strength (17.772 ksi) leads us to expect a finite life and hence a fatigue analysis based on Von Mises stress values\(^2\) was carried out. This showed that crack initiation was likely to occur at about 28,000 cycles. In fact, a sample of this design failed on the bump table at about 15,000 cycles. Contours of Log\(_{10}\) values of life in cycles are shown in Figure 5. The 0.5" design brings the stresses down to such an extent (about 10,000 times) that fatigue failure is not to be worried about.

<table>
<thead>
<tr>
<th>Analysis type</th>
<th>Result</th>
<th>0.25&quot; design</th>
<th>0.5&quot; design</th>
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</thead>
<tbody>
<tr>
<td>Linear static at static load</td>
<td>Deformation</td>
<td>0.27x10(^{-3}) inch</td>
<td>0.22x10(^{-3}) inch</td>
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<td></td>
<td>Max. Principal stress</td>
<td>6.209 ksi</td>
<td>3.384 ksi</td>
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<tr>
<td></td>
<td>Max. shear stress</td>
<td>3.104 ksi</td>
<td>1.692 ksi</td>
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<tr>
<td></td>
<td>Von Mises stress</td>
<td>5.674 ksi</td>
<td>3.229 ksi</td>
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<tr>
<td>Linear static at &quot;3g&quot; impact (bump test)</td>
<td>Deformation</td>
<td>0.62x10(^{-3}) inch</td>
<td>0.66x10(^{-3}) inch</td>
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<tr>
<td></td>
<td>Max. Principal stress</td>
<td>18.626 ksi</td>
<td>10.151 ksi</td>
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<tr>
<td></td>
<td>Max. shear stress</td>
<td>9.313 ksi</td>
<td>5.075 ksi</td>
</tr>
<tr>
<td></td>
<td>Von Mises stress (S)</td>
<td>17.023 ksi</td>
<td>9.688 ksi</td>
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<tr>
<td>Fatigue failure (crack initiation)</td>
<td>Min. life to fatigue failure (N)</td>
<td>0.28818x10(^{6}) cycles</td>
<td>0.16161x10(^{6}) cycles</td>
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<tr>
<td></td>
<td>Log(_{10})(N)</td>
<td>4.46</td>
<td>8.21</td>
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</tbody>
</table>

CONCLUSIONS

In comparison with the stationary application design of this hermetic reciprocating compressor, internal mounting brackets as well as compressor feet were modified to meet the additional tough requirements posed by the all-terrain mobile application. The final design has passed all the field and laboratory tests and is capable of meeting the tough impact, vibration and environmental requirements of the mobile application.

REFERENCES

1. EMRC, User's Manuals for NISA, Display III and Endure, 1993, Troy, MI, USA.
Figure 1. Sketch of the internal mounting bracket showing design alternatives

Figure 2. Deformation under static load for the 0.25" design
Figure 3. Von Mises stress contours for the 0.25" design at "3g" impact load (static analysis)

Figure 4. Von Mises stress contours for the 0.5" design at "3g" impact load (static analysis)
Figure 5. Log-life contours for the 0.25" design from fatigue analysis

Figure 6. Log-life contours for the 0.5" design from fatigue analysis