Compressor Discharge Line Stress Analysis Utilizing Finite Element Methods

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ABSTRACT

Finite Element Methods have been utilized in the stress analysis of a refrigerant compressor internal discharge gas line failure problem. F.E.M. provided a useful guide to solving the problem quickly without requiring extensive laboratory test work.

Using F.E.M. to help solve this problem was a two step process. First discharge line stresses were calculated for a list of displacements the compressor subassembly was assumed to make inside the shell during shipment. As it turned out, only one of the possible displacements investigated produced calculated stresses high enough at the known point of failure to indicate a possible problem.

Once the failure promoting displacement had been defined, several discharge line modifications were analyzed using this displacement in an attempt to reduce the stress level in the problem area. After F.E.M. analysis of possible design modifications had yielded a satisfactory design for this worst case, the new design was rechecked with all the assumed shipping displacements to ensure that a new problem had not been created elsewhere in the discharge line.

METHOD

Using F.E.M. to help solve this problem developed into a two step process as the work progressed. The work was begun by compiling a list of possible motions the compressor subassembly could make relative to the compressor shell during shipment. Some types of motions would be restrained by the internal compressor mounting scheme more than others, but this effect was not considered when listing possible motions.

Samples of the relative displacements investigated could be described as pure vertical motion, pure horizontal motion, a combination of vertical and horizontal motion, and a combination of horizontal or vertical or both with the subassembly tipping from vertical. The horizontal motion or tipping could be in any of four directions with respect to the compressor mounting feet.

Layout drawings of the compressor were then used to determine the maximum relative displacements possible between the ends of the discharge line for each of the motions listed. The compressor drawings were also used to determine the spatial coordinates of the nodes for the F.E.M. analysis. The configuration of the discharge line dictated that this would be a 3 dimensional analysis problem. A sampling of the subassembly motions that were studied are shown in Table 1.

The input data for the computer program was assembled using these node coordinates and the relative displacements between the two ends of the discharge line. To input the displacement data to the computer program, one end of the discharge line was assumed to be fixed while the other end was displaced the maximum relative displacement possible for each motion studied.

A computer run was made using each of the assumed displacements with a printer plot included in each to summarize the output. This helped greatly to reduce the pages and pages of printed results into a concise form that could be easily compared from the desirable approach. Instead, Finite Element Methods were used to do this work and did indeed provide a solution to the problem quickly.

INTRODUCTION

A mature product with many years of successful shipment and service demonstrated a very high failure rate due to shipping damage when shipped to the Mid East. It was felt that the long trip on board ship and subsequent travel over unimproved roads contributed to the damage. Examination of these shipping failures showed the problem to be a failure of the compressor internal discharge gas line just before it exits through the hermetic compressor shell. This compressor had the motor-housing subassembly mounted on isolation springs inside the shell. The discharge line contained a shock loop for added flexibility but some type of subassembly motion was causing the line to fail.

This problem needed a solution quickly. Solving this problem with empirical data gathered through many hours of laboratory testing did not look like
one run to the next. The same consecutive element numbering scheme was used for each problem with the final end of each element always joined to the initial end of the next element.

The initial list of possible displacements included many more possibilities than are shown in Table 1. After the first series of computer runs had been made, many motions could be eliminated because no high stresses were indicated or two types of motion would give the same magnitude of stress values. An example of identical stresses would be equal magnitude displacements upward and downward. This elimination was done in an effort to reduce the computer time required for the remaining work.

The displacement that was judged to be the worst case and that was used for the design variation studies is the last one shown in Table 1. This was with the compressor subassembly moving .2 inches downward and .3 inches horizontally towards the terminal box.

At this point various design changes could be evaluated quickly to see if they might reduce the stress levels. Simple changes such as varying the thickness of the discharge line require changing only one input number. The effect of reduced subassembly travel can be seen. The evaluation process would require more time if the shape of the lines were changed. This would involve data for new node coordinates and possibly more elements.

One feature of the computer program that was essential to this work was the ability to input as data specified displacements for selected nodes. For this work, the top end of the tubing was always fixed while the node at the opposite end was given the specified input displacement. This displacement was the relative displacement between the ends of the tubing that corresponded to the assumed motions of the compressor subassembly relative to the compressor shell.

The program has provision for a data check run which provides a printed listing of all numbers input as data and a plotted output of the assembled element grid to aid in checking node coordinates. The element grid is plotted as projected onto the 3 planes formed by the system coordinate axes chosen for the problem. A pictorial view is also plotted. An example of the pictorial data check plot for the compressor discharge line analyzed in this work is shown in Figure 1. Utilizing this plot of input data greatly reduces the time required to check the input data and eliminates subtle mistakes.

The program uses the input displacements instead of input loads to calculate the corresponding displacements of all other nodes which in turn are used to find the member forces at the nodes of each element. These member forces are then used along with the physical properties of each element to calculate stresses within each element.

Stresses are calculated at each end of an element and at a specified number of equally spaced internal points. Five stress numbers are calculated and
FIGURE 2 Stress plot for relative displacement of .2" downward, standard tube.

printed for each stress calculation point. Axial, torsional, shearing and bending stresses are calculated and combined to give an equivalent stress at each stress calculation point. By doing this, the several varying stress numbers can be combined into one number that might be compared from one loading condition to another.

Each of the equivalent stress components is calculated as detailed below:

a. Maximum stress due to axial forces.
   \[ S_a = \frac{P}{2\pi R t} \]
   P = Axial force
   R = Tube radius to the center of the wall thickness.
   t = Tube wall thickness.

b. Maximum stress due to bending moments.
   \[ S_b = \frac{M C}{I} \]
   M = Vector sum of the bending moments about the element principle axis.
   C = R + 1/2 t
   I = \( \frac{1}{4} R^4 t \)

c. Average stress due to torsional moments.
   \[ S_t = \frac{T}{2\pi A} \]
   T = Torsional moment
   A = \( \frac{2\pi R^2}{4} \)

d. Average stress due to beam shearing forces.
   \[ S_v = \frac{V}{4\pi R t} \]
   V = Vector sum of the shear forces in the element principle axis directions.

These four stress values are combined into an equivalent stress value using the Maximum Distortion Energy theory of failure.

FIGURE 3 Stress plot for relative displacement of .4" lean towards terminal box, standard tube.
In this case, without an internal pressure in the discharge line, 

\[ S_e = \left( S_x^2 + S_y^2 - S_{xy}^2 \right)^{1/2} \]

\( S_x \) = Normal stress acting on a plane perpendicular to the axial direction in the finite element.
\( S_y \) = Normal stress acting on a plane parallel to the axial direction in the finite element.
\( S_{xy} \) = Shear stress acting in the \( S_x \) and \( S_y \) plane.

In this case, without an internal pressure in the discharge line, \( S_y = 0 \). The expression for the equivalent stress then reduces to:

\[ S_e = \left( S_x^2 + S_{xy}^2 \right)^{1/2} \]

\( S_x = S_a + S_b \)
\( S_{xy} = S_t + S_s \)

The same idea is followed with the bending and axial stress. The bending moments about the two element axis are combined into one positive moment for the stress calculation. This yields a maximum bending stress number \( S_b \), however, the location on the circumference of the element is not found. The absolute value of the axial force \( S_a \) is used to determine the stress due to axial forces.

When the stress values are combined to give the equivalent stress, the maximum normal stress and the maximum shear stress are assumed to apply at the same point in the material. This is not always the case, but doing so yields stress data.
about the worst possible case and is the conservative and expedient approach.

The post processor section of the computer program is a tremendous help in condensing the stress information generated by the program and placing it in a manageable form. This is accomplished with a line printer plot of the calculated equivalent stress values that is printed at the conclusion of each run. This plot displays the calculated equivalent stress values versus position along the discharge line. The equivalent stress values are scaled with the largest value in any one run to the size of the paper and plotted as shown in Figure 2.

Each line in the printer plot represents the value of the calculated equivalent stress at a specific point along the discharge line. In this work each element was approximately one inch long with stress calculations performed at each end of the element and at two equally spaced calculation points for each element. The combined plot of each stress point versus position along the line provides a concise visual representation of the calculated equivalent stress distribution.

Here is where the benefits of consistent node and element numbering pay off. With element lengths approximately the same, the many computer runs that need to be analyzed can be quickly compared. However, the possible differences in absolute magnitude between any two printer plots must be kept in mind.

RESULTS

Reducing the problem to a study of one type of motion did make the design work progress quickly. The several design alternates looked at included an increase in the tubing wall thickness at the failure point and the length of this thicker walled portion of the discharge line. As each design was run the effect of the change was studied at the failure point as well as throughout the remainder of the discharge line.

The magnitude of the reduction in the calculated maximum stress can be seen in Figures 4 and 5. The calculated equivalent stress at the discharge line termination, which is represented at the top of Figures 4 and 5, was reduced 25 percent with the revised line configuration. The stress level at the other end of the line, which is represented at the bottom of Figures 4 and 5, increased about one percent. An increase in stress levels of 1 to 10 percent was seen throughout the remaining portion of the discharge line. The higher stress levels occur because the strengthened portion of the tube reduces the flexibility at the discharge line termination increasing the displacements and stress levels throughout the remainder of the line.

With a possible discharge line design defined, the original set of assumed displacements was run again as a check to ensure that a new problem had not been created at another point in the discharge line. Small increases in stress on the order of 4 to 10 percent were seen in the rerun of the other displacements.

The approach of using a set of assumed motions worked well in this problem because one of the assumed motions showed very high calculated stresses at the known point of failure. However, it would not have been cumbersome to evaluate design changes using several displacements. Even if several displacements required evaluation, only a maximum of twelve numbers in the input data would have to be changed to evaluate another possible displacement once the physical data for the problem is loaded in the computer.

REFERENCES

