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STRESS ANALYSIS OF SUSPENSION SPRINGS AND DISCHARGE LOOP UNDER IMPACT CONDITIONS FOR HERMETIC COMPRESSORS USING F E M

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ABSTRACT

The transport worthiness of suspension springs and discharge loop in hermetic compressors under impact conditions has been analysed by analytical and computational simulation. The equations represent static behaviour to consider the maximum stresses arising from the impact loads. The problem is divided into two parts. The discharge loop is analysed by linear one-dimensional finite elements in 3-D space. Fixed end boundary conditions are applied at the Shell end of the loop and Load conditions are applied at the Discharge Muffler end. The loads are transmitted by the springs to the pump and motor assembly and evaluated analytically. The discharge loop is considered as an additional spring and the spring constant is determined using the FEM. Using the analytical formulae and FEM the failure loads and stresses are calculated for two types of loop geometries and two types of mounting systems. However the method can be extended to other combinations of loop geometry and springs.

INTRODUCTION

In a low side hermetic compressor, the motor and pump assembly hereafter referred to as mass, is mounted on springs either extension or compression and the high pressure discharge gas leaves the shell by means of a long tube. This long tube which acts as a spring connected to the mass, takes the impact loads as well as running loads. While the running loads and oscillations are predictable, the impact loads occur due to unintentional dropping of the compressor with or without packing material and due to bad transport infrastructure It has been observed that failure often occurs in the springs and/or in the discharge loop, rendering the compressor unusable causing considerable loss to the manufacturer.

In the design of Hermetic Compressors different spring mountings and discharge loop geometries can be considered depending on the vibration analysis of the mass and the space available in the shell. Another factor for the loop is the heat transfer to the lubricant oil. It is very difficult to experiment with the combination of springs and discharge loop and is expensive to manufacture compressors and test them for variety of impact loads and transport conditions. A number of models have been proposed for analysing the discharge tube (1,2). However, these models do not take account of interaction between the suspension springs and the discharge tube. Hence a method is proposed in which the discharge loop and the suspension system can be analysed together.

Analysis of two designs of discharge tube and mounting systems are presented. The results show the advantages of this analysis and also point to the best design. Maximum failure stresses are calculated for the discharge loop, which most likely breaksat the shell end.

COMPUTATIONAL MODEL

The computational procedure is carried out in two parts. First the discharge tube is considered to be made of 1-dimensional linear elements, with varying cross section, moment of inertia and material properties, and the tube occupies 3-dimensional space (3). One end of the tube is connected to the shell and the other end is connected to the Discharge Muffler. The Muffler is considered as a rigid body along with the Pump, Motor, Cylinder Head etc. The loads from the shell due to impact are transmitted to the springs and to the mass consecutively and the tube acts as a spring between the Shell and the mass.

In the Finite Element Analysis of the tube, we have to solve the equation,

$$[K\rho](u) = (F)$$

Where,

(F) = Force Vector consisting of Loads and Moments.
$= (F_x, F_y, F_z, M_x, M_y, M_z)$

$(u) = (u, v, w, \emptyset_x, \emptyset_y, \emptyset_z)$

By taking a polynomial for the shape functions as a function of (4), with the appropriate boundery conditions we get

$$\Phi_1(\xi) = 1 - 3(\xi/L)^2 + 2(\xi/L)^3 \qquad \Phi_3(\xi) = 3(\xi/L)^2 - 2(\xi/L)^3$$

$$\Phi_2(\xi) = -2L(\xi/L)^2 + L(\xi/L)^3 \qquad \Phi_4(\xi) = -L(\xi/L)^2 + L(\xi/L)^3$$

For the present problem where the principle direction is ξ along the tube and ζ and η are normal to ξ, the nodal displacements and rotations in ζ and η directions are related to the shape functions and the stiffness matrix is given by:

$$[K\rho] = EI \int [\Phi'']^T [\Phi''] d\xi$$

Integrating the above, the stiffness matrix contribution for displacement (v) and rotation ($\emptyset_\eta$) normal to ξ is,

$$EI \begin{bmatrix} 12/L^3 & 6/L^2 & -12/L^3 & 6/L^2 \\ 6/L^2 & 4/L & -6/L^2 & 2/L \\ -12/L^3 & -6/L^2 & 12/L^3 & -6/L^2 \\ 6/L^2 & 2/L & -6/L^2 & 4/L \end{bmatrix}$$

For displacement (u) and rotation($\emptyset_\xi$) along the tube.

$$[K\rho] = \frac{EA}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \text{ for } u$$

and $\left[K_P\right] = \frac{GJ}{L}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}$ for ϕ_ξ

Where E is the modulus of elasticity, I the moment of inertia, A the area of crosssection and L the length of the element.

The Element Stiffness Matrix for each element is calculated using the above matrices and transformed into Element Stiffness matrix in the Global sense by considering the Direction Cosines of each Element with reference to a Global coordinate system.

$$\left[K_P\right]^{GLOBAL} = \left[T\right]^T \left[K\right]^{ELEMENT}\left[T\right]$$

Where (T) is the transformation matrix consisting of Direction cosines of the local co-ordinate system with reference to the global co-ordinate system.

Boundary And External Conditions:

The discharge tube is fixed at the shell end and all the displacements and rotations are constrained at the point where the tube is attached to the shell (though it may extend further inside the compressor). At the Discharge Muffler end the forces and moments transferred from the springs through the mass are applied according to the local and global coordinate system at that end.

Deflections and Stresses:

In the first run displacements at discharge muffler end are calculated for different forces and moments applied in the X, Y, Z directions and the spring stiffness matrix is determined. According to the model described for the distribution of load in the springs and by applying the constraints for the motion of the mass within the shell the maximum forces/moments are obtained. These forces are again applied at the Muffler end of the tube to obtain the deflection of the tube, and the stresses developed at the fixed end are calculated and the failure criteria is applied.

Spring System Analysis:

To calculate the distribution of impact forces among the springs and the discharge tube, the procedure outlined in (5) has been used, treating the problem as a static deflection one. The discharge tube is also treated as a spring.

The stiffness matrices of the springs for displacements at spring ends are calculated using the standard formulae (6). For the loop, this matrix is derived using the FEM model. These stiffness matrices are transformed and added to yield the global stiffness matrix, acting on the displacements of the centre of gravity of the mass.

The final equation to be solved is of the form: $\left[K\right]\left\{Sg\right\} = \left\{f\right\}$

Where: (K) is the global spring stiffness matrix {Sg} the displacement vector of the centre of gravity and {f}, the impact forces and moments acting at the compressor centre of gravity. The system is solved and the stiffness matrix of each spring is used to arrive at the force vector experienced by each of the springs.

This force vector is used to calculate the maximum stress developed in the springs and the discharge tube.

RESULTS

Two types of Loop-suspension system which are already in use were considered. The models are shown in figures 1 and 2. In the first model the mass is suspended by extension springs and the discharge tube is submerged in oil at the bottom and the tube is of varying cross sectional area. In the second model the springs are of compression type and the discharge tube goes above the motor and outlet is still located at the bottom of the shell. In the second model it should be noted (Fig.2) that a part of the tube is fixed to the shell and hence that part is not considered to act like a spring.

The impact forces experienced by the compressor are evaluated for the case, when the maximum possible displacements occur. Impact loads in all three directions and combinations of these directions have been considered that lead to the maximum displacements. For these forces, the stresses in the springs and the discharge tube have been calculated.

PARAMETER	DESIGN 1	DESIGN 2
Max. load possible for design 1	300 N (67.4 lbf)	300 N (67.4 lbf)
Max, shear stress in springs	1743 MPa (253 kpsi)	634 MPa (92 kpsi)
Max. stress in discharge tube	1067 MPa (155 kpsi)	155 MPa (22.5 kpsi)
Max. load possible for design 2	——	500 N (112.3 lbf)
Max. shear stress in springs	——	1260 MPa (182 kpsi)
Max. stress in discharge tube	——	849 MPa (123 kpsi)

The maximum possible load is considerably higher for design 2 and the stress level at this higher load are less than the stress levels in design 1 at its lower maximum possible load.

From this analysis and Figs. 3 and 4 it can be seen that the load carrying capacity of the second design is much larger compared to the first design.

CONCLUSIONS

Though only two cases are presented, the model can handle any combination of tube geometry and spring system. Efforts are being made to make the package robust and interactive so that the geometry, boundary conditions and loads can be changed easily to obtain an optimum design. Further applications include analysis of the discharge tube - mounting system for compressors used on mobile platforms.

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Fig. 1

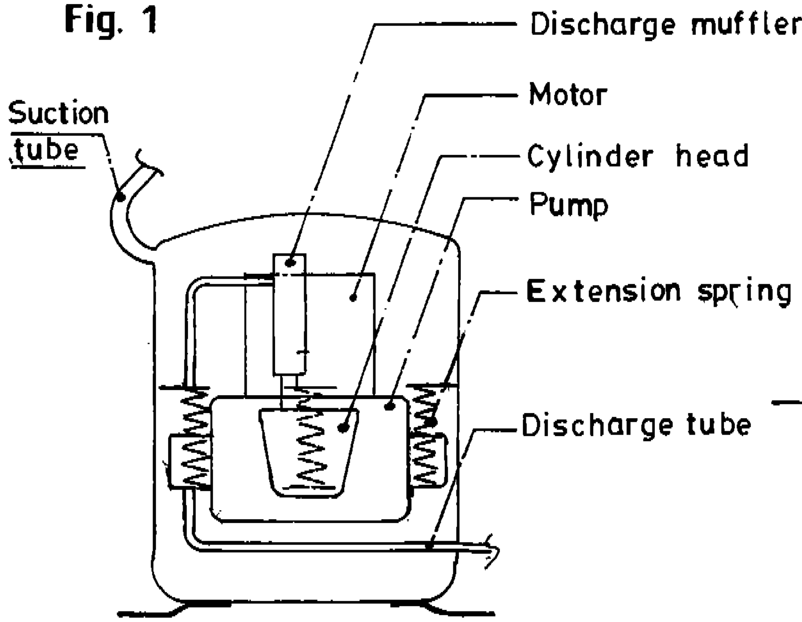

Fig. 2

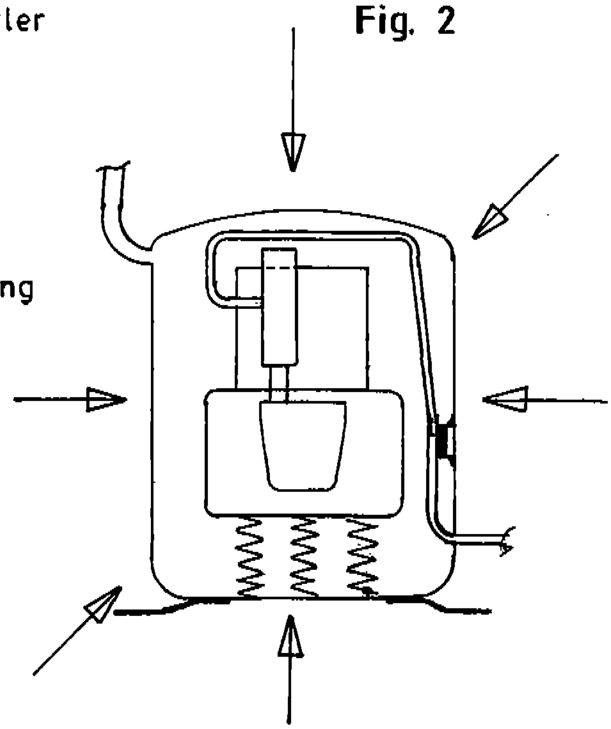

FIG. 3 Des. 1

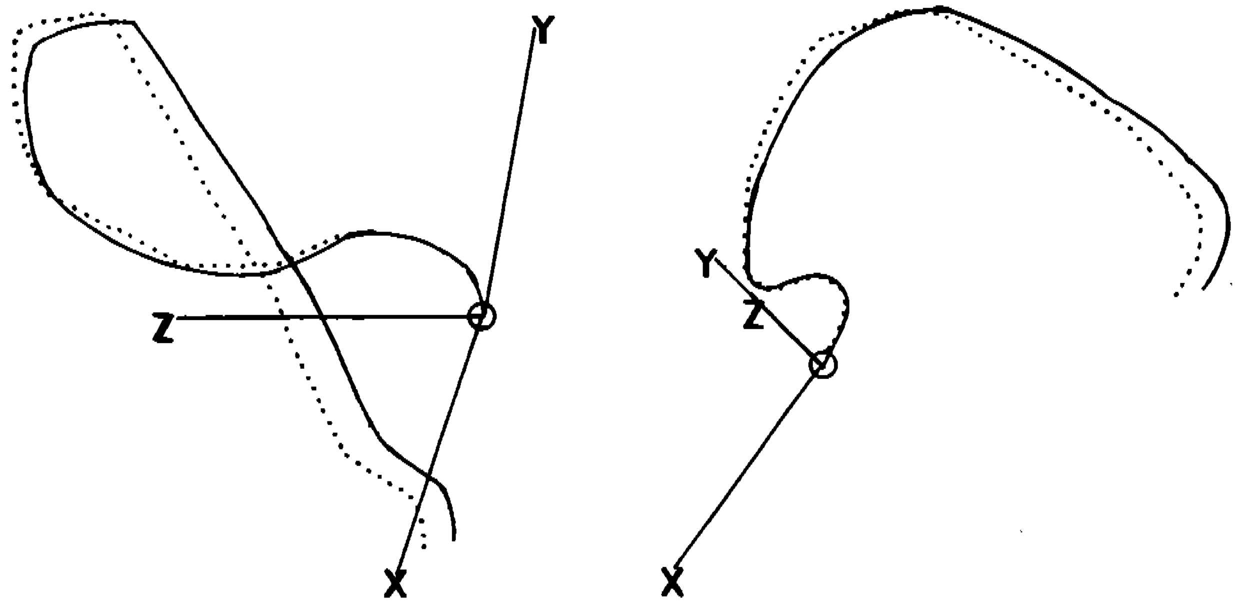

FIG. 4 Des. 2