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S. Akella
Shrlram Refrigeration Industried Ltd.

N. J. Rao
Shrlram Refrigeration Industried Ltd.

E. V. Venugopal
Shrlram Refrigeration Industried Ltd.

K. Venkateswarlu
Shrlram Refrigeration Industried Ltd.

FINITE ELEMENT ANALYSIS OF COMPRESSOR VALVE DYNAMICS

AKELLA S, RAO N J, VENUGOPAL E V, VENKATESWARLU K.

SHRIRAM REFRIGERATION INDUSTRIES LTD.
BALANAGAR, HYDERABAD - INDIA.

ABSTRACT

A 9 degrees of freedom triangular plate element is compared with the conforming and high precision Cowper element. The suction and discharge valves of a room airconditioning compressor are modelled; the natural frequencies mode shapes, displacement and stresses have been evaluated. Static load, dynamic load and boundary conditions on the valves are discussed.

INTRODUCTION

The suction and discharge processes of a hermetically sealed compressor are controlled by reed type valves. The thermodynamic performance, noise performance and overall efficiency of the compressor is affected by the valve dynamics. Correct values of natural frequencies and mode shapes are required for noise analysis, vibration analysis and modal analysis for compressor simulation model. The valve displacement helps in the calculation of gas flow area. Fatigue life due to pulsating gas pressure can be evaluated by the stress calculation of valves. Valve mechanics and design is discussed by Soedel (3), the triangular plate element of Cowper (1) is applied by Hamilton (2) for valve analysis. Futakawa (4) and Lal (5) have also applied finite element technique to valve analysis and design. Dusil (6) has evaluated the fatigue life of valves due to valve impact on seat.

In the present paper the 9 degrees of freedom (d.o.f) triangular plate element is evaluated with Cowper element so that valve analysis with more nodes and elements is possible. The dynamic pressure pulsations were considered for the valve response and stress analysis in addition to the static load. The suction and discharge valves of a 1.5 ton hermetic compressor are analysed.

THEORY

The Cowper element is based on a quintic polynomial and 18. d.o.f per element. The stiffness matrix, mass matrix, load vector and transformation matrices are given by Cowper (1) and Hamilton (2).

Hamilton (2) also mentioned about the 9 d.o.f element but has not fully developed and tested it. This element has a quadratric polynomial as the basis function.

$$W(x,y) = a0 + a1x + a2y + a3x^2 + a4xy + a5\,y^2 + a6x^3 + a7xy^2 + a8y^3$$

Nodal d.o.f

Cowper	9 d.o.f.
W	W
$W_{,x}$	$W_{,x}$
$W_{,y}$	$W_{,y}$
$W_{,xx}$	
$W_{,xy}$	
$W_{,yy}$	

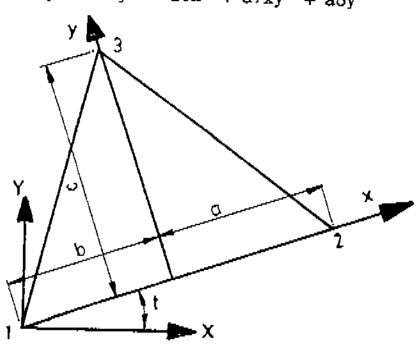

Fig 1. TRIANGULAR PLATE ELEMENT

The suction valve is clamped firmly between the valve plate and the cylinder housing face and bends like a cantilever until it hits the valve stop. For further deflection the free end of the suction valve gets pinned against the valve stop. The discharge valve is backed by a spring support, the valve stops all along the surface against the valve stop during flexing. Some points of the valve might come in contact with the valve stop before others, which will alter the boundary conditions, such changes in boundary conditions are not considered in this analysis. The discharge valve is rivetted at the base and is assumed as a clamped end.

Valve loading

The valves face the port locations on the valve plate which are shown in Figures 2 and 3. The under pressure, difference between cylinder pressure and suction line pressure, is about 25 psi, and the over pressure, difference between cylinder pressure and discharge line pressure, is about 40 psi (7). The differences are due to pressure losses across the valve ports, valves, mufflers and tubing. Generally, the maximum pressure difference across the valve is upto 3 psi and depends on the valve configuration. This pressure difference acts as a static load at the points corresponding to the ports. Apart from the static load, the cylinder has pressure pulsations which are recorded using a dynamic pressure transducer (7). The peak to peak pressure pulsations at 47.5, 95, 143, 285 and 427 Hz are 2.1, 0.95, 0.47, 0.18 and 0.01 psi. These pressure pulsations cause dynamic loading on the valve. The responses and stresses of the suction valve are calculated for the static load and for each of the dynamic loads. Invoking the linearity principal the responses and stresses are added to obtain the total stress. The pulsations in the suction line are not considered, these pulsations are to be vectorially added to the cylinder pressure pulsations to get the net dynamic load on the valve. For the case of the discharge valve, the valve sits against the valve seat under static load and on further loading the valve is compressed against the valve seat.

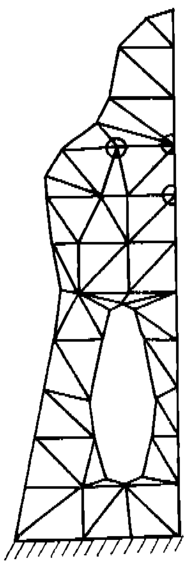

Fig 2 SUCTION VALVE

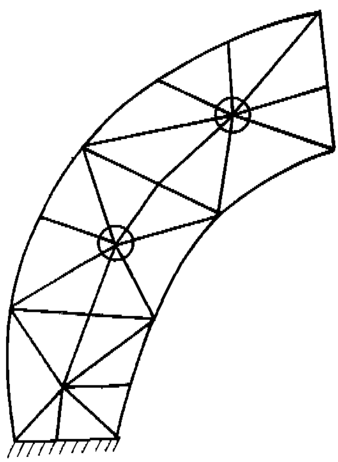

Fig 3 DISCHARGE VALVE

In this element, the complete quadratric polynomial is not taken; x^2y term is left out, one of the ways of including it is taking the co-efficient of xy^2 and x^2y term same, imposing an additional constraint on the element. This element has also been tried but is not discussed as it behaves similar to the basic 9 d.o.f element. The nodal degrees of freedom are W, $W_{,x}$ and $W_{,y}$. The transformation matrix [T1] which relates the nodal degrees of freedom $\{W1\}$ and the co-efficient vector $\{A\}$ is:

$$
\{W1\} = \begin{bmatrix}
1 & -b & 0 & -b^2 & 0 & 0 & -b^3 & 0 & 0 \\
0 & 1 & 0 & -2b & 0 & 0 & 3b^2 & 0 & 0 \\
0 & 0 & 1 & 0 & -b & 0 & 0 & 0 & 0 \\
1 & a & 0 & a^2 & 0 & 0 & a^3 & 0 & 0 \\
0 & 1 & 0 & 2a & 0 & 0 & 3a^2 & 0 & 0 \\
0 & 0 & 1 & 0 & a & 0 & 0 & 0 & 0 \\
1 & 0 & c & 0 & 0 & c^2 & 0 & 0 & c^3 \\
0 & 1 & 0 & 0 & c & 0 & 0 & c^2 & 0 \\
0 & 0 & 1 & 0 & 0 & 2c & 0 & 0 & 3c^2
\end{bmatrix} \{A\}.
$$

Where a, b and c are the lengths as shown in Figure 1. The rotation matrix [R] consists of three diagonal matrices [R1] and relates the local degrees of freedom W1 to the global degrees of freedom W .

$$
[R1] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \text{Cos } t & \text{Sin } t \\ 0 & -\text{Sin } t & \text{Cos } t \end{bmatrix}, \quad [R] = \begin{bmatrix} R1 & 0 & 0 \\ 0 & R1 & 0 \\ 0 & 0 & R1 \end{bmatrix}, \quad \{W1\} = [R]\{W\}.
$$

Where t is angle between local and global coordinates. The procedure for obtaining mass and stiffness matrices and load vector is similar to the Cowper element (1). In Cowper element stresses are directly obtained from nodal curvatures. However, since curvatures are not nodal degrees of freedom for a 9 d.o.f element, stresses have to be derived from nodal slopes and displacements.

$$
\begin{Bmatrix} W_{,xx} \\ W_{,xy} \\ W_{,yy} \end{Bmatrix} = [RS]\,[B]\,[T1]\,[R]\,\{W\},
$$

$$
[RS] = \begin{bmatrix} \text{Cos}^2 t & 2\,\text{Cos } t\,\text{Sin } t & \text{Sin}^2 t \\ -\text{Sin } t\,\text{Cos } t & \text{Cos}^2 t - \text{Sin}^2 t & \text{Cos } t\,\text{Sin } t \\ \text{Sin}^2 t & -2\,\text{Sin } t\,\text{Cos } t & \text{Cos}^2 t \end{bmatrix},
$$

$$
[B] = \begin{bmatrix} 0 & 0 & 0 & 2 & 0 & 0 & 6x & 0 & 2y & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 2y & 2x & 0 \\ 0 & 0 & 0 & 0 & 0 & 2 & 0 & 2x & 0 & 6y \end{bmatrix}.
$$

[RS] [B] [T1] [R] and $\{W\}$ are evaluated for an element which has the node under study. It is thus possible to obtain the nodal curvatures from different elements, which share the node.

Valve configuration

The suction and discharge valves of the hermetic compressor are shown in Figures 2 and 3 respectively. Both the valves are symmetric in shape, boundary conditions and loading. Hence, only one half of the valves is used for the finite element model. For convergence test, finite element models with different nodes and elements were tried. The models shown in Figures 2 and 3 have the highest number of elements. The suction valve has 48 nodes, 56 elements, 288 d.o.f with Cowper element and 144 d.o.f with the 9 d.o.f element. The discharge valve has 19 nodes, 22 elements, 114 d.o.f with Cowper element and 57 d.o.f with the 9 d.o.f element.

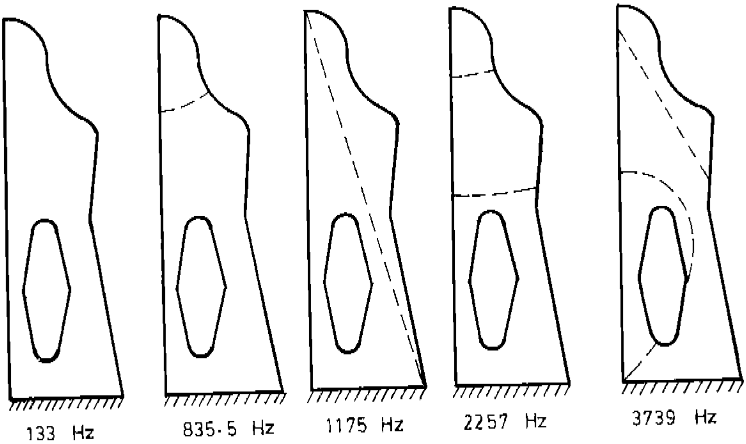

Fig 4. SUCTION VALVE MODE SHAPES

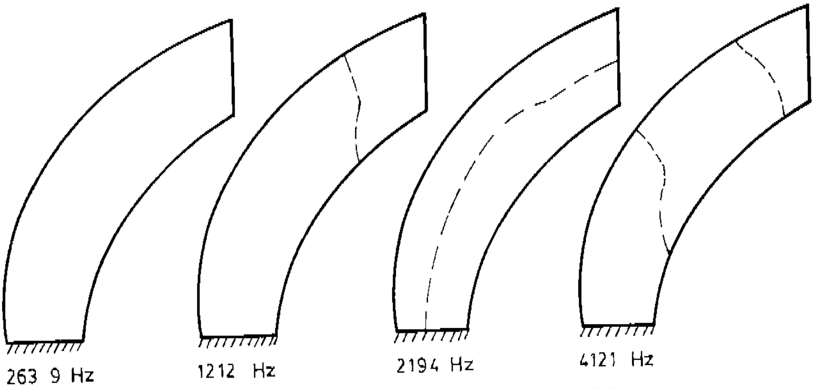

Fig 5. DISCHARGE VALVE MODE SHAPES

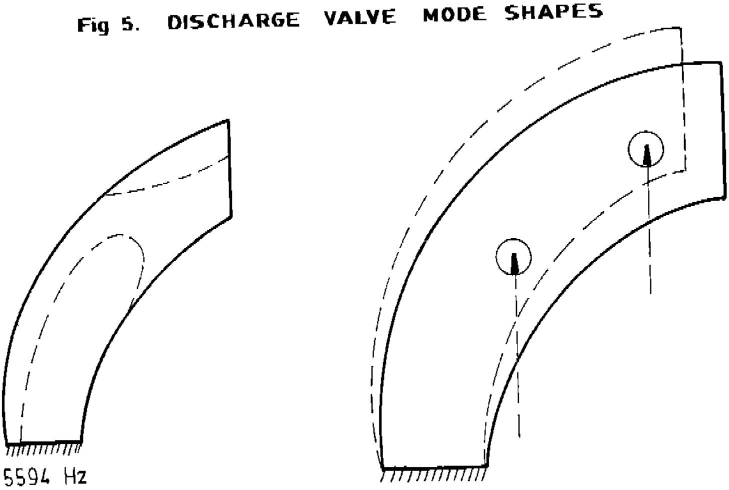

Fig 6. DISCHARGE VALVE DEFLECTION

Results and Discussion

Cowper element has curvatures as nodal d.o.f, hence, it can incorporate both fixed and pinned boundary conditions. Also, stresses can be directly calculated from the nodal displacement vector. However, in the case of a 9 d.o.f element, pinned end cannot be completely defined. Also, stresses have to be derived from nodal displacement vector using transformations. Both the elements were compared with illustrations given by Cowper (1), for static and dynamic performance.

Both suction and discharge valves behave like cantilevered plates, the natural frequencies and mode shapes are shown in Figures 4 and 5. The first, second and the fourth modes are longitudinal modes with one, two and three lateral nodal lines respectively. The third mode is a lateral mode, with one longitudinal node and the fifth mode is a combined mode. The natural frequencies and mode shapes of the suction valve in fixed - pinned configuration are not shown.

The discharge valve reaches the valve stop when the pressure difference

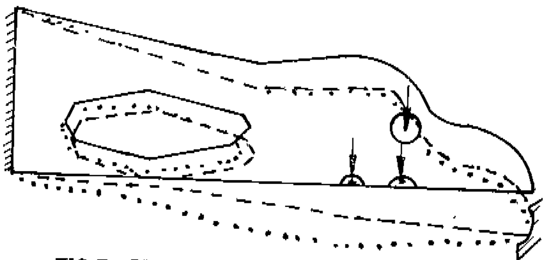

FIG.7. SUCTION VALVE DEFLECTION

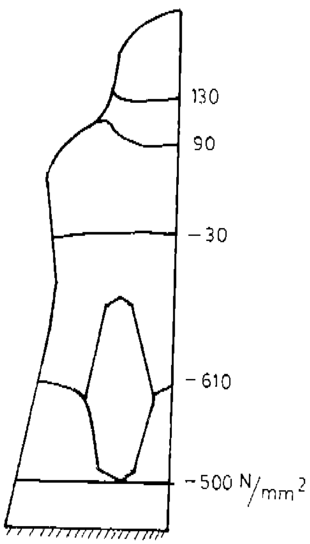

Fig 8. STRESS CONTOURS OF SUCTION VALVE

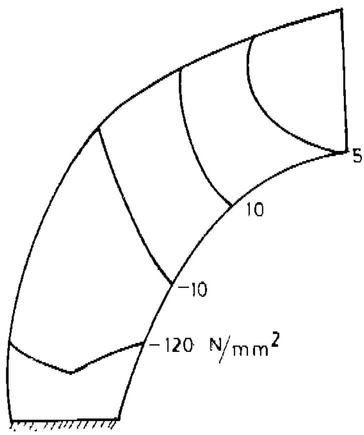

Fig 9. STRESS CONTOURS OF DISCHARGE VALVE

across the valve is 1.4 psi without the spring support as shown in Figure 6 and when the pressure difference across the valve is 1.8 psi with the spring support. After reaching the valve stop no further flexural, stresses are developed, the direct compressive stresses are negligible ($2N/mm^2$) compared to the flexural stresses ($120\ N/mm^2$) as shown in Figure 9. In the case of a suction valve the valve first bends as a cantilever till it touches the valve stop when the differential pressure across the valve is 0.5 psi, at this point the boundary condition of the valve changes, a further deflection occurs under a static pressure of 2 psi across the valve. The deflections as a cantilever and as a fixed pinned plate are shown in Figure 7. For the stress analysis, dynamic load at 47.5 Hz and 95 Hz is considered and the combined stress contours are shown in Figure 8. The dynamic stresses due to 149, 285 and 427 Hz were very low and were neglected. The main stresses of the suction valve are due to static load and the dynamic load contributes only 15 per cent of the total stress.

The tensile stress of the valve material given by Dusil is $1860\ N/mm^2$ and the elastic limit is $1420\ N/mm^2$. From the SN curve for the given thickness (0.3825 mm) of material for 10^9 cycles of life, 2.3% probability of fracture is at a stress level of $\pm\ 770\ N/mm^2$. The discharge valve is found to be clearly safe. However, the suction valve has high stress levels ($600\ N/mm^2$) at the base of discharge port opening hole. This might be shifted to failure due to improper barrelling and tumbling (6). However, the high stress zone is away from the zone of valve impact against the valve stop. Fracture analysis of returned compressors have shown 50% failure at the high stress zone. The stresses for both valves were calculated along the longitudinal and lateral directions and also along the principal directions at each of the node points. The longitudinal stresses are discussed and plotted as these are easier to visualise than principal directions and are higher than lateral stresses due to more flexing along the length. The stress contours and shift from compression to tensile stresses are understood from super position of valve modes.

CONCLUSIONS

The 9 d.o.f element is useful for evaluating natural frequencies and mode shapes with increased nodes, as condensation is not possible. However, for stress analysis Cowper element is recommended. The valve dynamic analysis are useful in design, analysis and quality review.

LIST OF REFERENCES

1. Cowper G R, et. al, 'A high precision triangular plate bending element' Aeronautical Report, LR-514 NRCC

2. Hamilton J F, 'Introduction to Finite Element Analysis of Vibrations and stresses in Gas Compressor Valves', Ray W Herrick Laboratories, Purdue University, 1980

3. Soedel W, 'Design and Mechanics of Compressor Valves', Ray W Herrick Laboratories, Purdue University, 1984.

4. Futakawa A, Namura K, 'Deformation and stresses of Refrigeration Compressor flexible ring valve', 1976 Purdue Conference, PP 195-202.

5. Lal D N, 'The use of Finite Element Method for stress Analysis of Compressor valves', 1978 Purdue Conference, PP 203-208.

6. Dusil R, Johnson B, 'Fatigue Fracture Behaviour of Impact Loaded Compressor Valves', 1978 Purdue Conference, PP 124-128.

7. Akella S, et. al., 'An Experimental Method for Design and Performance of Hermetic Compressor', Vol. 2, 1986, Purdue Conference, PP 428-443.