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THE DYNAMIC BEHAVIOUR OF HALF-ANNULAR
VALVE REEDS IN RECIPROCATING COMPRESSORS

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ABSTRACT

The Finite Element Method (F.E.M.) was used to study the static and dynamic behaviour of half-annular reeds with and without a backing plate. Static displacements and stresses were evaluated and the reed shapes and frequencies were derived when vibrating freely. The convergence of the analytical solutions was examined. The predicted mode shapes and natural frequencies were in good agreement with experimental results obtained by time-averaged laser holography. Dynamic displacement and stress patterns were calculated when a reed was subjected to a variable pressure difference during the suction or discharge phases of a compressor cycle. Significant secondary effects were considered.

INTRODUCTION

To be adequate a mathematical model of a reciprocating compressor must include a simulation of the automatic suction and discharge valves. For many compressors the moving element of the valve is a flexible reed covering one or more ports. To protect the reed from over-stressing the permitted lift is limited either by a point stop near the tip or by a backing plate. The resulting boundary conditions at impact on the stop at opening (or on the seat at closing) are non-linear and complicate the mathematical model required to simulate the dynamic behaviour of the reed.

The displacement and stresses in reeds of the cantilever type due to bending, both under static and dynamic conditions, have been evaluated by the present authors, using the F.E.M. (1,2,3,4,5): stresses due to impact have not yet been considered. The suite of computer programs developed(3) provides also for (a) the generation of the mesh automatically (b) the minimisation of the bandwidth of the finite element matrices and (c) a visual display of dynamic displacements.

Typical grid sizes employed to analyse a half-annular reed are shown in Figure 1. When this type of reed is employed in a discharge valve a backing plate is usually fitted in order to limit the lift and hence the bending stresses. Elson et al (7) described a semi-analytical, one dimensional model to study the dynamic displacement at the tip of this shape of reed.
STATIC ANALYSIS

The suitability of the finite element model for static analyses had been verified by investigating the central displacement, edge and central moments of a clamped or simply supported plate of rectangular or square shape subjected to uniformly distributed pressure. (6, 8).

The static displacement and stress patterns of a half-annular reed with a backing plate have been examined by the first author (6). Although the gas pressure loading on a reed is distributed over the area of the valve port, it was assumed initially that the drag force due to the pressure difference across the reed in the area of the valve port could be considered as concentrated at the port centre. This simplifying assumption impugns the accuracy of the predicted static (and dynamic) stresses in the area of the port. However, it does not severely affect the predicted static (and dynamic) displacements over the whole reed nor the static (and dynamic) stresses away from the port area (St. Venant's principle). The finite element model developed can account for a distributed and consistent loading in both the static and dynamic analysis, but much greater computer resources would be required, particularly in the dynamic case. (Alternatively in a dynamic analysis, a correction factor can be introduced (3, 6)).

The grid size which was employed for the static analysis is shown in Figure 1(c). Since the exact nature of the boundary conditions for the reed at contact with the backing plate is not known a-priori, the effects on the reed behaviour of changing the boundary conditions were investigated.

Figure 2 illustrates static displacement patterns of the reed under different boundary conditions and confirms that the stiffness of the reed increases as it wraps about the backing plate.

FREE VIBRATIONS

Initially the Chalda sand method was used to estimate the natural frequencies and mode shapes of reeds. This technique does not convey information regarding the points of maximum displacement (anti-nodes) and the sand tends to alter the dynamic characteristics of the thin reeds.

Mode shapes
Time-averaged laser holography (1, 6) was used to find some of the mode shapes of a thick (2.26 mm) half-annular reed similar to the thin (0.20 mm) reed analysed. The thickness of the test specimen was relatively large in order to avoid instability problems which may arise.
during the application of the time-averaged laser holography. In order to obtain almost perfect clamping conditions at the roots, the thick reed was machined from a much thicker steel plate: the unmachined part was then clamped in a fixture and the thick reed part was vibrated by a piezoelectric crystal driven by an oscillator, via a charge amplifier (1,6). (Such driving crystals could alter the dynamic characteristics of a thin reed although an acoustic (but unpleasantly noisy) horn could have been used).

Plate 1
Fundamental mode shape, 661 Hz.

Plate 2
1 Radial mode shape, 1742 Hz.

Plate 3
2 Radial mode shape, 3462 Hz.

Plate 4
1 Circumferential mode shape, 4000 Hz.

Plates 1 - 4 depict examples of the laser holograms obtained (some included by courtesy of Dr. A.J. Waddell). When the reed vibrated freely at 4000 Hz, the circumferential node did not have the same centre as the reed. This suggested that the thick reed was not perfectly clamped; a phenomenon examined by Kennedy (9).

The position of the applied driving force had an effect on the observed mode shapes of the thick reed. Even if a force was applied at a node, normally assumed to remain stationary, the thick reed vibrated at slightly higher than its natural frequency because extra strain energy is required when a form other than the most natural form of vibration takes place. The
mode shape became apparent only after the force was moved away from the node.

Natural frequencies

Experimental and analytical values of natural frequencies of the thick reed are compared in Figure 3. The small differences were a consequence of imperfect clamping in the experiment. This resulted in analytical values (where ideal clamping conditions were assumed) always being slightly larger than experimental results. Rapid but non-monotonic convergence was evident. The assumed state of displacement in the triangular elements employed yields an overestimate of the element stiffness (1). To satisfy internal but not boundary compatibility in the elements results in a decrease of stiffness. The two effects nearly cancel and fast but non-monotonic convergence results. There was also some randomness in the accuracy of the predicted values of natural frequencies because a particular grid size might describe some mode shapes better than others (1). Antisymmetric grids generally represent antisymmetric modes better than symmetric modes, while symmetric grids represent symmetric modes better (1).

Figure 4 illustrates the extent of convergence of the predicted natural frequencies of the thin reed over a range of size of grid. The effect of differently orientated grids, but having the same number of degrees of freedom (d.o.f.), could affect the predicted natural frequencies by 5 - 10%.

Since the thin and the thick reed had the same geometry with the exception of thickness, the predicted natural frequencies closely followed the scaling law:

\[
\frac{f_2}{f_1} = \frac{t_2}{t_1}, \quad \text{where}
\]

\[ f : \text{the frequency, and} \]

\[ t : \text{the thickness of each reed.} \]
The effect of imperfect clamping on the predicted natural frequencies could be allowed for by including additional elements at the root of a reed, as illustrated in Figure 5 (1,3). Such elements, when attributed with a Young's modulus of 5 x 10^9 - 1 x 10^10 Kg/mm^2, allowed for a reduction of about 10% in the first three natural frequencies of the steel reed.

Comparisons between the analytical and experimental mode shapes of the thick reed are shown in Figure 6. The small differences, particularly at the 4000 Hz (symmetric) mode shape, are due to imperfect clamping conditions in the experiment (9). Allowance for this imperfection was not made in the present analysis because of the higher demand which would be imposed on computer resources. Figure 7 illustrates the predicted mode shapes of the thin reed.

**Response to Dynamic Loading**

The grid sizes used to study the response to dynamic loading of the half-annular thin reed are shown in Figures 1(a) and 1(b). Fewer d.o.f. were used than in the static analysis (Fig. 1c) in order to achieve a reduction in computer resources. Conditionally or unconditionally stable direct integration (step-by-step) schemes can be employed based either on the Newark - β or the Wilson - β method (1,3,6).

**Figure 7.** Predicted natural frequencies and mode shapes of the half-annular thin reed.
The dynamic displacement at the tip of the thin reed under typical compressor discharge conditions without a point stop or backing plate to limit the permitted lift is illustrated in Figure 8. A conditionally and unconditionally stable integration scheme was used (1). Figure 8 demonstrates that relatively large time steps can be employed with such unconditionally stable integration schemes (when the reed is between the seat and stop) without loss of accuracy and with saving of computer time. The finite element model used needs small integration time steps to deal with the non-linear boundary conditions of a reed when it is in touch with the valve stop or seat (3,6).

Figure 9 shows the dynamic response of the reed tip without restriction to the lift when an impulse load of 1 N was applied during one time interval (1,6). The response is plotted for a range of values of the damping coefficient, $C$. The reed vibrated mainly in its first mode. Higher frequency vibrations disappeared as the allowance for damping was increased. When the damping coefficient was $C = 0.01$, no oscillation occurred because the allowance was more than the critical value of damping which was $C = 0.005$.

The predicted motion of a radial cross-section of the reed at the centre of a port with only a point stop at the tip, under typical compressor discharge conditions, is shown in Figure 10. The large predicted displacements result in large stresses in the reed and demonstrate the necessity of a backing plate. The reed overshot the permitted lift at the tip, reversed a little due to increase in its stiffness following tip contact and then reached its maximum displacement. The motion of the reed depends on the particular...
pressure-time history acting across the valve which in turn depends on the operating conditions of the compressor.

The motion of a radial cross section of the thin reed with a backing plate in place under typical compressor discharge conditions using a finite element grid of 27 d.o.f. (Figure 1(a)) is shown in Figure 11. The reed was arrested by the backing plate at node 4 and 13 and dwelt in contact with it. The reed failed to touch the stop at node 22 but oscillated there while the other nodes were at rest on the backing plate.

However it was considered that there may have been an over-estimation of the stiffness of the reed at the inner radius as a consequence of using a relatively crude grid with only 27 d.o.f. (Fig.1(a)). Node 22 contacted the backing plate when a finer grid with 84 d.o.f. was employed and the dynamic loading was distributed to two points inside the area over the valve port (nodes 25 and 49, Fig. 1(b)). Node 22, on the crude grid (Fig. 1(a)) became node 70 on the finer grid (Fig. 1(b)).

The effect of the position of the backing plate on reed behaviour, using the finer grid is illustrated in Figure 12.

Figure 13 shows dynamic stresses evaluated under typical compressor discharge conditions for the half-annular thin reed with the backing plate in place. The maximum pressure difference across the reed during the discharge phase of the compressor cycle was 30 lbf/in² (6). At the line of symmetry of the reed, transverse stresses \( \sigma_y \) were much larger than longitudinal stresses \( \sigma_x \) due to the geometry of the
A large change of the predicted values of both bending stresses at points A and B occurred when the reed reached the backing plate at point A, Figure 13. The accuracy of the predicted values of dynamic stresses at point A, which is inside the port area, may be dubious since the applied force is assumed concentrated at the centre of the port and not distributed over the whole port area.

SECONDARY EFFECTS

Oil stiction of a valve reed at its seat (or stop) can influence the behaviour of the reed by affecting the effective impact velocity at contact with stop (or seat). Brown and Pringle (10,11) developed a semi-analytical model to simulate the oil stiction phenomenon. Stiction was found to be mainly dependent on the width of the valve seat, reed thickness, flexibility, surface finish, oil viscosity and the rate of change of gas pressure across the reed.

Allowance for the oil stiction effect was made in the finite element model by inclusion of a time delay factor \(\tau\) which retarded the departure of the reed from the seat when opening or stop when closing. This delay factor could be estimated by the model developed by Brown et al (10). The delay was stated (10) to be almost inversely proportional to compressor speed: there is a higher rate of change of pressure difference across the valve as compressor speed is increased. However, the oil stiction effect tends to be of more significance at high compressor speeds because of an increase in the ratio of the delay time, \(\tau\), to the duration of the suction or discharge phase of the compressor cycle. During this delay, the pressure difference is building up across the closed reed, resulting in a larger initial accelerating force on the reed when it ultimately breaks off the seat (or stop). Consequently the effective impact velocity at stop (or seat) is increased.

A prediction of the effect of oil stiction on the displacement and impact velocities of a reed (with a backing plate) is shown in Figure 14, where a 10 to 25% increase in predicted impact velocities would result and thus accelerate any failure due to impact fatigue. The delay in commencement of valve closure due to the stiction between the reed and the surface area of the backing plate, besides increasing the effective impact velocity on the valve seat, may also cause loss of compressor capacity by back flow as a consequence of late valve closure. To reduce the effect of oil stiction small holes or grooves in the seat and/or the backing plate of a reed are sometimes added.

A "squish" effect due to a layer of gas and oil trapped between the reed and the backing plate (or seat) decelerates the reed just before impact. MacLaren et al (12) demonstrated this effect with ring plate valves. Elson et al (7) included damping forces to account for this effect in a model for a half-annular reed. From "squeeze film" lubrication theory, these forces are inversely proportional to the cube of the small distance between the reed and the backing plate. The squish effect is more pronounced in discharge valves where there is a backing plate. Neglect or underestimation of this effect results in overestimation of the predicted effective impact velocities. The combined oil stiction and squish effects tend to counter one another as regards their influence on the effective impact velocities.
CONCLUSIONS

A finite element model was applied to predict the static and dynamic behaviour of a half-annular reed.

The results of the static analysis were not particularly meaningful since the boundary conditions of the reed while in operation are not known a-priori. Investigation of several combinations of boundary conditions (as suggested by Futakawa et al (13)) did not provide reliable information on valve performance and reed durability. Hence a dynamic analysis of the reed behaviour is deemed to be necessary despite the complexity of the model and the great computer resources required.

The Finite Element Model developed gave good agreement between the predicted natural frequencies and mode shapes and those measured by time-averaged laser holography. The predicted results under free and forced dynamic conditions were sensitive to the number of degrees of freedom employed in the model. Hence an analysis of convergence is necessary.

Comparison of analytical and experimental results of displacements and stresses in cantilever type valve reeds under forced dynamic conditions (when a prescribed pressure-time history was applied) have been reported previously by the authors (2,3,4,6). The predicted reed displacements and stresses for the half-annular reed under operating conditions were not verified experimentally in this study but appeared to be intuitively correct.

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REFERENCES


