International Compressor Engineering Conference   School of Mechanical Engineering

1974

An Impact Test Rig for Annular Plate Valve Models

D. A. Forbes
University of Strathclyde

J. Mitchell
University of Strathclyde

Follow this and additional works at: https://docs.lib.purdue.edu/icec

AN IMPACT TEST RIG FOR ANNULAR PLATE VALVE MODELS

Donald A. Forbes, Lecturer
University of Strathclyde, Glasgow, Scotland

James Mitchell, Technical Manager
Scottish Stamping and Engineering Co. Ltd., Ayr, Scotland

GENERAL

Self acting or automatic valves are one of the most critical and least durable components in reciprocating compressors and their quality can greatly affect the economy, performance and reliability of a compressor. Although considerable thought has been given by designers to the behaviour of valves in reciprocating compressors, much of the design of this component remains empirical. While simple design procedures based on beam theory are useful from a comparative standpoint, the design of prototypes, far removed from existing practice, requires a fundamental knowledge of valve plate mechanics for successful extrapolation. Increased valve life is one of the aims of successful valve design and an improved understanding of valve stress behaviour is fundamental·to the achievement of this end.

The experimental study of valves may be approached in two ways; firstly, with the valve inside the compressor so that all service features are retained but difficulty with instrumentation is experienced, or, secondly, with the valve represented by a simpler, larger scale model which may be easily instrumented but necessarily depicts non-service behaviour. Both approaches are considered in this paper.

SELF-ACTING RING PLATE VALVES

Static Considerations

The behaviour of self acting ring plate valves may be conveniently represented by valve plate displacement and pressure difference plotted to a base of crank angle. Figure 1, by MacLaren and Kerr (1) shows such a diagram of experimental and theoretical valve behaviour. Two facets of behaviour, worthy of analysis from a stressing aspect are apparent, namely, static pressure difference across both valves and dynamic effects at opening and closure of the valves.

Theoretical studies of static uniform pressure loading on annular plates are well established (2), (3) and (4), but confirmation by experimental techniques for practical valve plate geometries requires some care. Nelson (5) analysed experimentally, using electrical resistance strain gauges, the static pressure behaviour of the valve plate (Figure 2) of a 9 hp Reavell compressor. The results, although indicating the correct trends, did not give close correlation with theory, mainly due to the small size of the valve plate, and the consequent difficulty of strain gauging small areas

with relatively large gauges. The difficulties associated with size were avoided by using a large scale model (outside diameter 11.5 inches) with all other parameters, such as spring stiffness, pre-compression, lift, etc, being established from a prototype by dimensional analysis. This model was of a convenient size for reliable strain gauging and testing statically by simplified means in an ordinary universal testing machine. It is shown in Appendix I that theoretically, circumferential ring loading could be substituted for pressure loading, the former being convenient to apply without the complication of fluid leaks or the possible lack of uniformity of the latter. Excellent agreement was achieved, between theory and experiment for a typical annular plate having an aspect ratio corresponding to ring plate valves, and is shown in Figure 3. As a result of this static analysis, some amendments to Shapiro's theoretical work were made which greatly improved correlation. Details of these amendments are reported in Appendix II. Figure 4 shows the theoretical pattern of radial and circumferential stresses for various aspect ratios, using Shapiro's amended theory.

Typical results by Nelson (5) for the Reavell valve plate were: for a static pressure across the valve plates of 112 lbf/in^2, the maximum radial stress was 11400 lbf/in^2 for a plate thickness of .036 inches and 4400 lbf/in^2 for a thickness of .058 inches.

Nelson stated that even if steel of poorest quality was considered, with an endurance limit of 30000 lbf/in^2 for 10^7 cycles, the measured and calculated stresses were well within this limit and failure by fatigue should not occur. However, fatigue data are usually derived from polished, rotating-bending test specimens and no account is taken of corrosion or stress concentrations which can considerably reduce the endurance limit.

It was concluded, from the initial work of Nelson and of the authors, that the simple static pressure differences existing across the valves under consideration were unlikely to be causes of failures and attention was focussed on dynamic effects.

Dynamic Considerations

Analytical approaches to the dynamic stress behaviour of annular plates fall into two main classes, namely, Energy solutions and Bessel function solutions. The first of these methods contains simplifying assumptions regarding plate deformation and is often further modified to simple

beam theory, as described by Berkman (4). Kinetic energy of the plate before impact is assumed to be converted on impact, without loss, to strain energy of bending of the plate. If the impact velocity is known, the plate deformation and consequent stresses can be obtained. Nelson, employing this analysis together with theoretical impact velocities derived by MacLaren and Kerr, obtained dynamic stresses four times greater in magnitude than the static stresses due to pressure difference across the valve.

The second analytical approach is based on classical plate theory combined with time dependent loading. However, even for plates with simplified loading and boundary conditions, the evaluation of natural frequencies by Weiner (6) required the solution of transcendental equations involving Bessel functions. Any advantage, therefore, accruing from the rigour of the solution is lost in the complexity of obtaining numerical values for specific examples. In view of the complications associated with analytical techniques, it was considered that an experimental approach would offer a better engineering solution to the problem and on the basis of simplicity, the rig shown in Figure 5 was designed.

IMPACT RIG

Figure 6 shows schematically the layout of the prototype impact rig and its operation. The annular plate representing a valve plate, initially rests on the inside and outside supports and is loaded statically by precompression of the spring via the concentric loading ring. Valve-lift is then obtained by raising the sleeve a predetermined amount on the central shaft and this "valve-open" condition is retained by the insertion of a pin through the sleeve and shaft. By releasing the pin, the valve plate model is "fired" against the support rings, thereby simulating rapid valve closure against its seat. A range of springs provides variation of valve loading while a range of support ring and loading ring heights and diameters allows variations of the valve plate aspect ratio and spring precompression. Holes bored at a series of levels on the central shaft provide variation in sleeve height and hence valve lift.

INSTRUMENTATION

Two types of recording devices were investigated in conjunction with piezo-electric gauges and electrical resistance strain gauges. The recording systems considered were, a Cathode Ray Oscilloscope with a camera and a 25-channel U-V recorder. The latter system proved more convenient in that the trace of the complete event, of preload and firing, could be easily obtained by starting the recorder just prior to removing the pin on the rig. The oscilloscope, apart from being restricted to the examination of only two gauges, required to be, for simplicity, self triggered by the incoming signal. Frequently it triggered on the first ½ wave of the impact, so missing the pin removal and plate acceleration periods. Continued use of the oscilloscope would have necessitated the design of an electronic device to start the oscilloscope trace just prior to the pin removal, and to avoid this, the U-V recorder system was adopted.

The dynamic strains in the plate during impact were measured using strain gauges bonded to the plate surface. Again a choice was made between piezo-electric gauges, having good frequency response and very high output, which would not require amplification, and electrical resistance strain gauges, with small gauge factors, which would require amplification. Electrical resistance strain gauges were chosen finally since the initial static preloads could be easily monitored as D.C. shifts in output, whereas the piezo-electric gauges which required the strain to change with time, were unsuitable for recording these preload conditions. Figure 7 shows the instrumentation finally adopted.

RESULTS

Figure 8 shows a typical trace from a piezo-electric gauge and while the pin removal, plate acceleration and impact are clearly shown, the static preload due to valve lift is not obtained.

Figure 9 shows a typical trace using the electrical resistance strain gauge set-up and the preload is clearly indicated. By comparing the magnitudes of the static preload and the maximum impact amplitude, an indication of the dynamic magnification is obtained while the frequency of the transient vibration is readily obtained from the timing lines on the recorder paper.

Initial tests were performed using a circular plate of diameter 11.5 inches whose dynamic characteristics could be readily calculated or obtained from tables. The experimentally measured natural frequency of the plate was within 12% of the theoretical value and measured dynamic stress amplitudes, 4 to 5 times greater than static levels, were in broad agreement with expected values.

Similar tests were repeated using annular plates and these gave the same general trace behaviour indicating satisfactory performance of the apparatus. A typical trace for an annular plate of aspect ratio b/a = .422, thickness = .175 inches is shown in Figure 10.

NON AXI-SYMMETRIC PROBLEMS

All the foregoing theory and experiment assumed complete axi-symmetry for ease of computation and strain gauge installation. An investigation such as impact of a "spinning coin" type would require large numbers of gauges on many diameters and this facet of the problem was not considered in these initial tests. An important topic which could easily be examined by the apparatus was the effect of solid foreign matter introduced between the plate and the support ring, so giving rise to a local stress concentration. This was investigated briefly using the set-up shown in Figure 11. Wire of varying diameters was introduced under the plate outside edge and a single strain gauge, mounted circumferentially on the plate top surface, directly above the inclusion, was monitored. Large amplitude traces were obtained indicating that foreign matter could be a source of crack initiations at the plate outside edge and might explain the radial cracking shown by Shapiro (3).

CONCLUSION

Initial tests with the simple impact rig described above, indicate that such an apparatus with slight modifications could be used to investigate a wide range of problems associated with the transient behaviour of annular plate valves.

BIBLIOGRAPHY

1. MacLaren, J. F. T., and Kerr, S. V., "An Analytical and Experimental Study of Self Acting Valves in a Reciprocating Air Compressor", Fluid Plant and Machinery Group, Inst. of Mechanical Engineers, October 1970.

2. Georgian, J. C., "Uniformly Loaded Circular Plates with a Central Hole and Both Edges Supported", A.S.M.E. Journal of Applied Mechanics, Vol. 24, March 1957, p. 306.

3. Shapiro, M. B., "Stress Calculation Procedure for Annular Valve Plates in Compressors", Translated from Khimichosko i Neftyanoe Mashinastroenie, No. 1, January 1965, p. 14 (in Chemical and Petroleum Engineering).

4. Berkman, B. A., "Self-Acting Valve Plate Strength in Compressors", Russian Engineering Journal, Vol. 44, Issue No. 10, 1964, p. 17.

5. Nelson, C. and Kerr, S. V., "Behaviour and Strength of Ring Plate Valves", Unpublished Report, Mechanical Engineering, University of Strathclyde, Glasgow, 1970.

6. Weiner, R. S., "Forced Axisymmetric Motions of Circular Elastic Plates", A.S.M.E. Journal of Applied Mechanics, Vol. 32, December 1965, p. 893.

APPENDIX I

From Georgian (2) the radial stress distribution may be computed for uniformly loaded annular plates simply supported at the inside and outside edges. For the same configuration, the radial stress corresponding to a concentric load located on the mean radius can be computed. Using a non-dimensional stress $\frac{\sigma t^2}{q\pi(a^2-b^2)}$ for the uniform pressure loading and $\frac{\sigma t^2}{P}$ for the concentric load case, the two stress systems may be compared noting that $q\pi(a^2-b^2)$ and P both represent the total load on the annular plates.

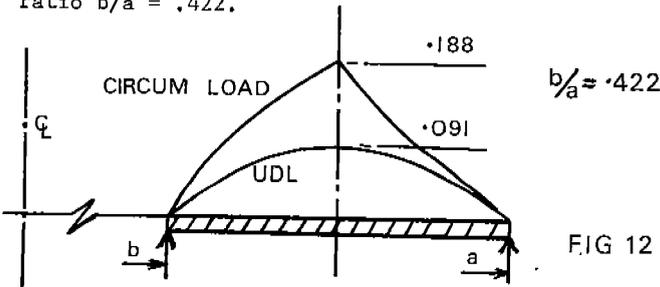Figure 12 shows the distribution of radial stress for the above two cases for a plate with an aspect ratio b/a = .422.

FIG 12

Considering also simple beam theory for beams of length (a - b), uniformly loaded and point loaded, by using the same non dimensional stresses, it can be shown that,

$$\frac{\sigma t^2}{q\pi(a^2-b^2)} = \frac{3\left(1-\frac{b}{a}\right)}{4\pi\left(1+\frac{b}{a}\right)}$$

for uniform pressure q and

$$\frac{\sigma t^2}{P} = \frac{3\left(1-\frac{b}{a}\right)}{2\pi\left(1+\frac{b}{a}\right)}$$

for central point load P.

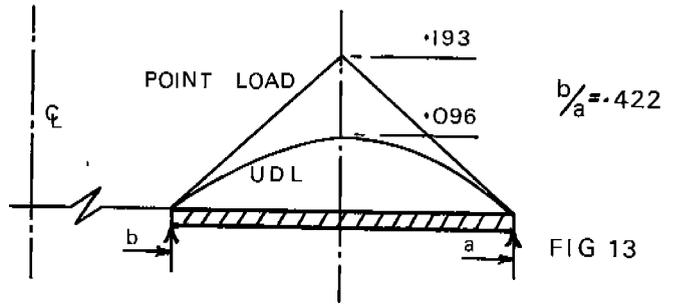The distributions of these stresses for a beam where b/a = .422 are shown in Figure 13.

FIG 13

The similarity between the two sets of results is obvious and indicates that considerable simplification is possible on both the experimental and theoretical aspects of the problem of disc valve stressing. Experiments may be carried out using circumferential loads and the maximum radial stress values which are obtained need only be halved to obtain the maximum radial stresses corresponding to uniform pressure loading. The difficulty associated with pressure loading and simple supports is therefore avoided.

APPENDIX II

Amendment to theoretical work by Shapiro (3)

Shapiro's theoretical study made use of the principle of superposition combining separate plate solutions to obtain the solution for a uniformly loaded annular plate. This approach and the theoretical distributions of circumferential stress for the intermediate and final plate loadings are shown in Figure 14.
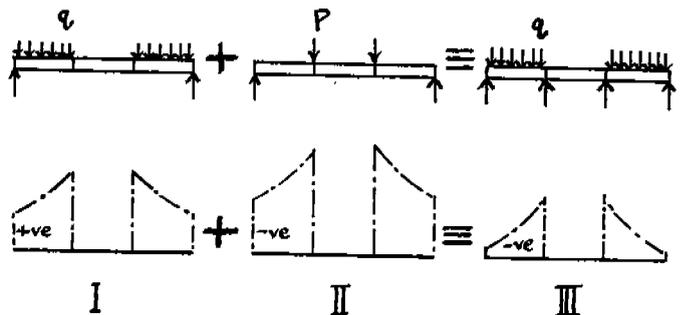
FIG 14

The first of these constituent stress distributions is given by Shapiro as,

$$\sigma_{\theta_{(I)}} = \frac{3q R^2}{2\delta^2}\left[\frac{(1+\nu)\ln X}{\alpha^2} + \frac{(3+\nu)}{4}\left(1+\frac{1}{\alpha^2}+\frac{1}{X^2}\right)\right.$$
$$\left. -\frac{(1+\nu)(X^2+1)}{(\alpha^2-1)X^2} - \frac{(1+3\nu)X^2}{4\alpha^2} - \frac{(1-\nu)}{\alpha^2}\right]$$

where $\alpha = \dfrac{a}{b}$, $X = \dfrac{r}{b}$

and it appears that a factor $\ln\alpha$ has been omitted from the third term within the brackets.

The amended equation reads,

$$\sigma_{\theta_{(I)}} = \frac{3q R^2}{2\delta^2}\left[\frac{(1+\nu)\ln X}{\alpha^2} + \frac{(3+\nu)}{4}\left(1+\frac{1}{\alpha^2}+\frac{1}{X^2}\right)\right.$$
$$\left. -\frac{(1+\nu)(X^2+1)}{(\alpha^2-1)X^2}\ln\alpha - \frac{(1+3\nu)X^2}{4\alpha^2} - \frac{(1-\nu)}{\alpha^2}\right]$$

where $\alpha = \dfrac{a}{b}$, $X = \dfrac{r}{b}$

with the resulting stress distributions as shown in Figure 15.
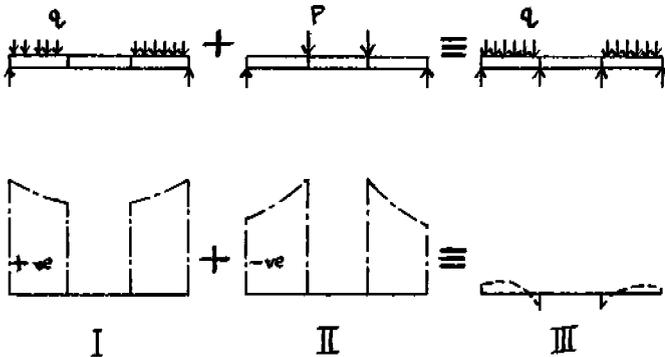
FIG 15

This amended final distribution of circumferential stress gave improved correlation with the authors' static experimental work.
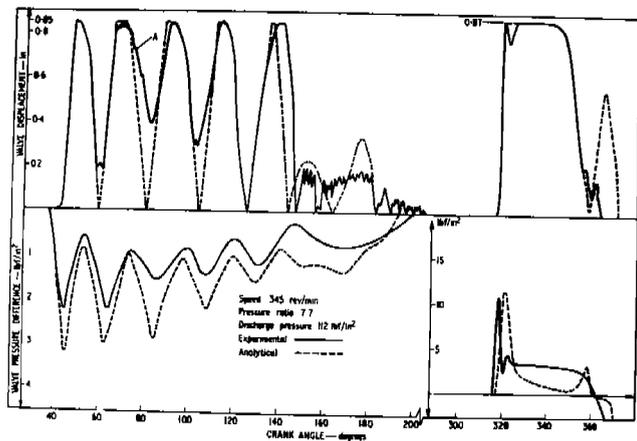
315

Fig.1   Valve plate displacement and pressure
difference plotted to a base of crank angle (ref.1)
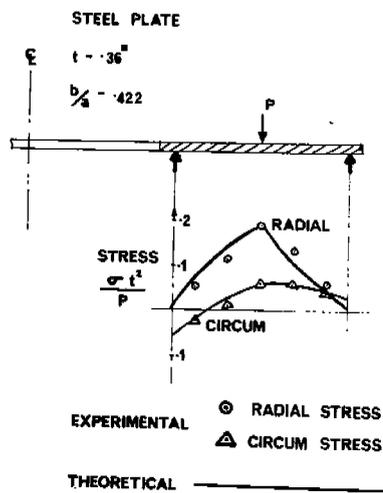

Fig.3   Comparison between theoretical and
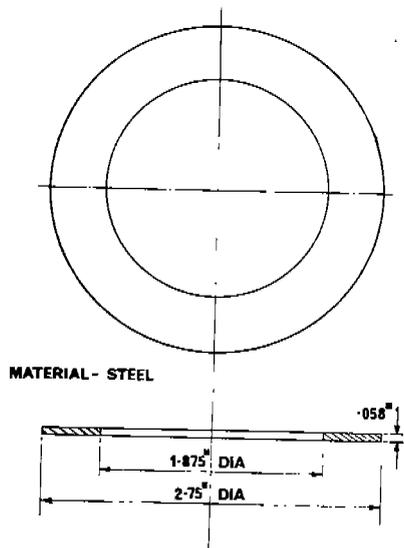experimental stresses for a typical annular plate
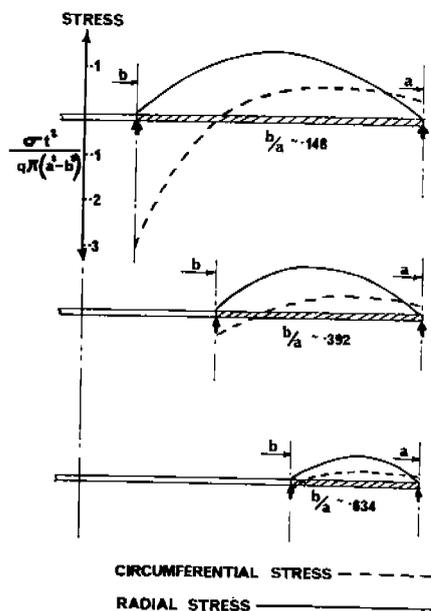

Fig.2   Valve plate detail (ref.5)

Fig.4   Typical distributions of stresses for
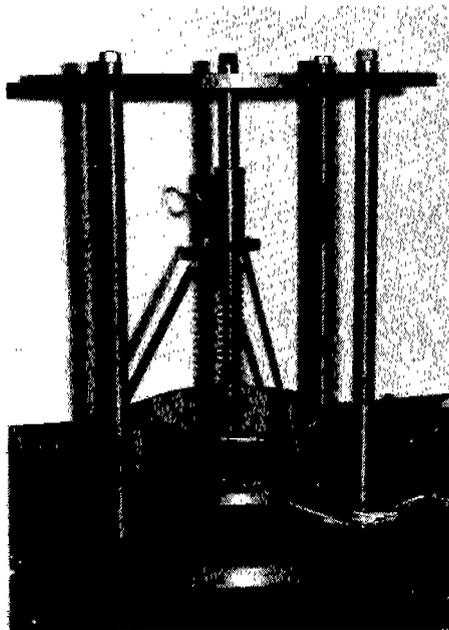uniformly loaded annular plates of various aspect
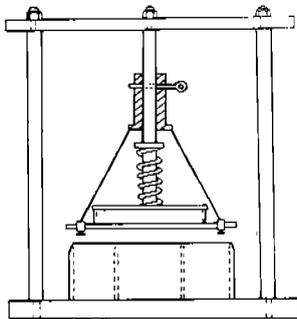ratios

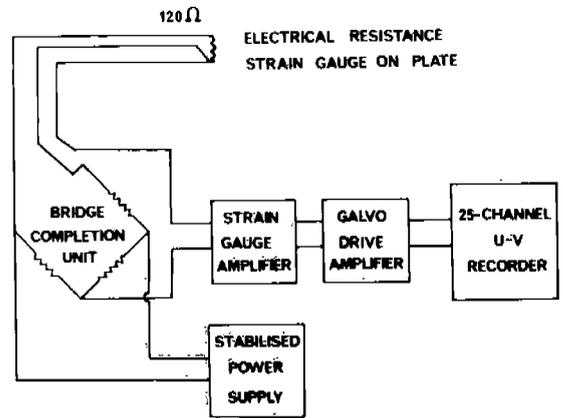Fig.5　Impact rig

Fig.6　Basic layout of impact rig

120 Ω

ELECTRICAL RESISTANCE
STRAIN GAUGE ON PLATE

BRIDGE COMPLETION UNIT

STRAIN GAUGE AMPLIFIER

GALVO DRIVE AMPLIFIER

25-CHANNEL U-V RECORDER

STABILISED POWER SUPPLY

Fig.7　Schematic layout of recording instrumentation

MAX. AMPLITUDE

A - STATIC PRE-LOAD.(NOT RECORDED)
B - PIN REMOVED.
C - PLATE STRIKES SUPPORTS.
D - FREE VIBRATION OF PLATE.
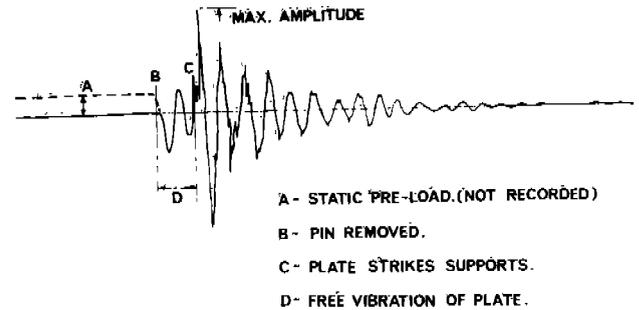
Fig.8　Typical trace using piezo-electric gauge

MAX AMPLITUDE

A - STATIC PRE-LOAD.
B - PIN REMOVED.
C - PLATE STRIKES SUPPORTS.
D - FREE VIBRATION OF PLATE.
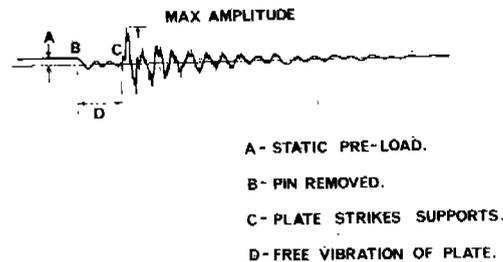
Fig.9　Typical trace from electrical resistance strain gauge

317

MAX AMPLITUDE

A ~ STATIC PRE-LOAD.
B ~ PIN REMOVED.
C ~ PLATE STRIKES SUPPORTS.
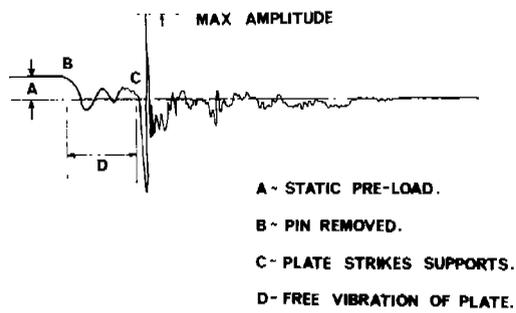D ~ FREE VIBRATION OF PLATE.

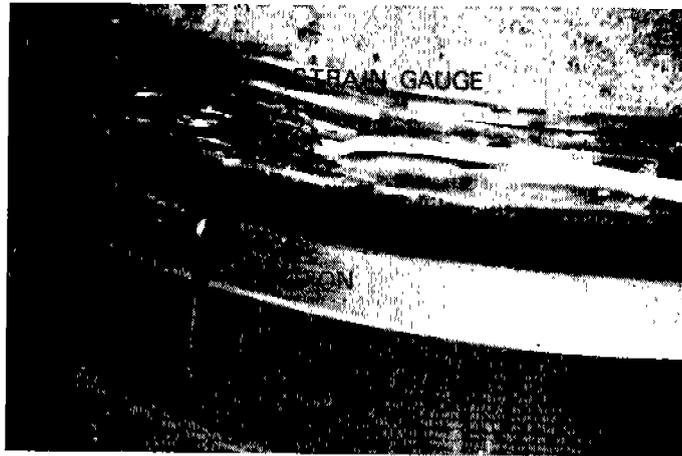Fig.10   Typical trace from annular plate impact

STRAIN GAUGE

Fig.11   Set-up with an inclusion under annular plate edge