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Increased Lift for Feather Valves by Elimination of Failures Caused by Impact

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ABSTRACT

The maximum lift that can be used in a feather valve for a particular application is limited by the valve failures that occur if the lift is too high. This limit has been increased by testing valves with a high lift and investigating the resulting failures. This has shown that the primary cause of failure is damage caused to the strip by impact of the strip on the guard.

As a result of this work, several design modifications have been made to the feather valve and these allow reliable operation at considerably higher lifts than were previously practical. This will, of course, have a direct influence on compressor performance.

INTRODUCTION

As part of our continual effort to upgrade compressor performance and reliability, we recently conducted an investigation of the possible failure modes of feather valves when operated at high lift and speed. The feather valve (Fig. 1) has been used with great success in a range of compressors for many years and at the commencement of the work described here we had a good understanding of the design guidelines to be followed to provide reliable service. However, we hoped to be able to improve the performance of compressors with feather valves and extend the range of application of feather valves by going outside these guidelines. Obviously this step would only be taken after adequate development and endurance testing.

As shown in Fig. 1, the strip in a feath. (or flapper) valve is both the sealing element and the spring and the design of this apparently simple component is greatly complicated by the many conflicting requirements, e.g. the strip thickness affects the spring force, the moving element mass, the stress in the strip at full lift and the differential pressure the valve can withstand.

Our intention was to develop a reliable valve with a higher lift than previously used for use in compressors running at around 900 rpm, a high speed for this valve size. To accomplish this we increased the lift to a point where valves in the test compressor in the Lab would fail every 50 to 100 hours and then set about determining the reasons for the failures.

It soon became apparent that there was no simple explanation for the failures, all calculated and measured stresses being well within the expected endurance limit of the strip material. We thus became involved in the extensive and varied work described here.

BASIC FEATHER VALVE STRESSES

While opening under the action of the pressure force, the feather valve strip behaves as a simply supported beam acted upon by a uniformly distributed load of varying magnitude. At full lift the strip rests against a guard which traditionally has a circular contour to provide the minimum stress for a given lift. In the high lift design that failed, the stress in the strip when the strip is bent to the radius of the guard is 26,700 psi, compared to an expected endurance limit for the strip material of at least 100,000 psi under "zero to maximum" stress conditions.

If the strip is assumed to be bent to the maximum lift by a uniform pressure load, the maximum stress is 35,200 psi and assuming the strip is vibrating in the fundamental mode with amplitude equal to the full lift, the peak stress is 36,100 psi.

ACTUAL BENDING STRESSES

It seemed probably that the actual stress in the strip would be higher than the basic stress given above due to the excitation of higher order vibrations. The effect of these was determined by strain gauge tests on operating valves. A typical strain gauge diagram is given in Fig. 2. The peak stress in the strip is 40 to 45% higher than that predicted by simple vibration or deflection under a uniform load. As shown in Fig. 2, the higher stress level is caused by a third order vibration excited in the strip during contact with the guard.

With a circular guard profile, the strip initially contacts the guard at the center and it was initially thought that the third order vibration was caused by the strip rebounding from the guard at the center while still moving towards the guard elsewhere. However, a detailed examination of stress and lift diagram (Fig. 2) suggests the following hypothesis (Figs. 3, 4).

Time A - B. The strip lifts under the action of the pressure force as a simply supported beam. The diagram shows only the first order being excited. In practice small amplitude, higher order vibrations can be excited, but these have no significant influence on the stress in the strip.

Time B. The center of the strip reaches the guard and rebounds slightly. The stress at this time σ_B is that given by a first order vibration of the strip to full lift at the center.

Time C. A point away from the center of the strip reaches the guard. At this time, the center of the strip is slightly off the guard and the stress is lower than it would be if the strip assumed the guard contour.

Time D - E. and F - G. The strip is held against the guard over a certain length at the center by the pressure force and the stress σ_D is that for the case when the strip is bent to the radius of curvature of the guard.

Time E - F. The pressure force is momentarily insufficient to hold the strip against the guard and it leaves the guard at points away from the center thus increasing the stress at the center.

Time G - H. At time G the pressure force becomes insufficient to hold the strip against the guard and it starts to leave. The center of the strip remains touching the guard until time H. All odd order vibration modes will be excited by this. The diagram shows only the 1st and 3rd modes as it is these that most strongly influences the stress in the strip. A small magnitude stress due to the 5th order is also discernable on the measured diagrams. The peak stress occurs at time H.

Time H - J. The valve closes with the 1st and 3rd order vibrations continuing to have the major influence on the stress in the strip.

A qualitative investigation of the vibration induced in the strip as it leaves the guard can be made by neglecting pressure forces and assuming that the strip is held in a circular arc and suddenly released. A Fourier analysis of the circular profile gives the amplitude of each order of vibration and the stress caused by each can then be evaluated as a function of time and summed. The results of the analysis considering just the first three odd orders is given in Fig. 5. The magnitude and form of the calculated stress diagram agree fairly well with those of the measured diagram, thus confirming that the above hypothesis is a possible explanation of the cause of the third order vibration.

The higher order vibrations can in theory be eliminated, whichever of the above two explanations of their cause is correct by shaping the guard so that the strip reaches the guard at all points simultaneously. The "sine contour" guard is an attempt to accomplish this and as shown in Fig. 6 it substantially reduces the stress due to the third order vibration.

MATERIAL PROPERTIES

The original strip material was 410 stainless steel heat treated to a hardness of 42-44 Rc which should have an endurance limit for "zero to maximum" loading of at least 100,000 psi. However, the fatigue test data available to us was obtained from conventional polished fatigue specimens. It seemed probable that the fatigue strength of rolled strip material was significantly less, especially as these strips had a fairly sharp edge and some obvious surface defects.

A fatigue test machine was therefore converted to stress simply supported strips by loading them at two points, thus creating a region of constant bending stress. To simulate conditions in the compressor, the strips were in an atmosphere of saturated air at 350°F. To our surprise the endurance limit of 410 SS strips was 145,000 psi and none of the 32 strips run at stresses of 135,000 psi or less failed. Thus this strip material has an excellent endurance limit with no evidence of quality control problems. Analysis of failed strips from the compressor tests had not shown metallurgical defects such as inclusions or surface flaws at the point of failure and it is concluded that simple fatigue is not a limiting factor in feather valve design.

During the course of this investigation many failed strips from the compressor were examined and in every case the initial failure was in the form of transgranular branched cracks with final failure by fatigue. This appeared to suggest that corrosion fatigue was a cause of failure, but as this material is used successfully in similar environments at higher stress levels and as similar failures were encountered in more corrosion resistant materials, this cause of failure was eventually ruled out.

IMPACT EFFECTS

It is commonly assumed among compressor designers that the impact velocity of the strip on the seat or guard has an important influence on the life of the strip. Several papers and books say that the velocity should be limited and many authorities have rules of thumb stating what the maximum velocity can be for a given type of valve.

A high impact velocity of the strip against the seat can deflect the strip into the gas passage causing high stresses and can also cause excessive wear in the sealing area. It is not evident how a high impact velocity into the guard causes strip failure, but from the type of failure and the relative values of the impact velocities in this case, it seemed likely that impact on the guard was more important than impact on the seat. Typical calculated and measured impact velocities on the guard were 18 to 20 ft./sec. which is in the range expected to give trouble.

Possible mechanisms by which high impact velocities can cause failures are:

a) Surface damage to the strip at the point of impact, caused directly by the impact, possibly in conjunction with small sliding movements of the strip on the guard. This effect might be increased by dirt or asperities on the guard surface.

b) High stresses caused by the strip inertia forcing the strip to conform to irregularities in the guard contour caused during manufacture or operation, or by dirt build up.

c) High stresses caused by stress waves produced in the strip by the impact. No such waves were measured by longitudinally or transversely mounted strain gauges, but this is not unexpected if the harmful waves are the plane stress waves caused by the deceleration during impact which have zero stress amplitude on the seat side of the strip, or high frequency bending or surface waves which might have too short a wavelength and too rapid a decay to be detected by the instrumentation used.

The magnitude of the compressive stress caused directly by the deceleration during impact can readily be calculated by conventional wave propagation methods. The stress wave diagram is shown in Fig. 7 and the results in Fig. 8. A more detailed analysis along the same lines has been published by Soedel (1).

It is evident that the transverse compressive stress is not sufficient to cause trouble even when combined with the longitudinal bending stress.

One of the features of stresses caused by impact is their high rate of strain. This can prevent a material from behaving in a ductile manner, e.g. at the tip of cracks and can cause a crack to propogate at a lower stress than would be required at lower rates of strain.

The work on an impact fatigue test rig by Svenzon (2) is of great interest. The rig tests showed that surface failures that lead to cracks can be caused in a steel strip by impact on a seat at velocities of about 25 ft./sec. The surprising finding is that the failures occur away from the zone of contact between strip and seat. This seems to prove that some form of stress wave causes the failure and that perhaps the lack of ductility at high rates of strain is important. The maximum stress calculated by a finite difference analysis in the test strips was 8,000 psi.

ENDURANCE TESTS IN A COMPRESSOR

During the course of this investigation, 29 different valve designs were tested in a compressor at 885 rpm. As stated earlier, the original valve lift had been chosen to cause early failures and the effect of design changes to the unsatisfactory design were examined. In each test, 4 valves each containing 6 strips, were tested until 3 strip failures or 4,000 hrs. operation was completed.

The results of the endurance tests showed that valve life could be improved by:

1) Using smaller (in all 3 dimensions) strips. This naturally requires the use of an increased number of strips and results in a more expensive valve with less flow area. Valves with 24 strips instead of 6 showed excellent reliability with the same static bending stress at full lift.

2) Shot peening the strips. It is necessary to experiment to determine the shot peening intensity necessary. The intensity initially recommended by one reputable supplier proved to be worthless. Peening with glass beads was also ineffective. The optimum shot peening increased valve life by a factor of 20 even though the endurance limit measured in the fatigue test machine was increased only 15,000 psi.

3) Using a sine contour guard. As described earlier this reduces the amplitude of the higher order vibration modes excited when the strip hits the guard. Use of the sine contour increased valve life by a factor of 8 while decreasing the measured maximum stress by 5%.

4) Using brass guards instead of cast iron ones. This increased strip life by a factor of 30.

5) Using a soft (non-metallic) "cushion" strip between the valve strip and the guard. In many thousands of operating hours with valves of this type, no valve strip failures have been encountered. Considerable work was however, required to find a suitable material for the cushion strips.

6) Reducing the lift. A reduction sufficient to decrease the measured stress in the strip by 30% increased the valve life by a factor of 10.

7) Using the optimum strip thickness. For the standard strip 5.7" long by .5" wide, the optimum thickness is .032". The strip life was halved by reducing the thickness to .028" or increasing it to .040". If the strip is too thin, the valve will not close early enough and failure may be caused by the resulting high impact velocity on the seat. If the strip is too thick, the simple bending stress is increased and this may cause failure.

During the testing, several different strip materials were tried including some developed specifically as premium flapper valve materials. The original choice of 410 SS proved to be as good as any.

Some initial development using non-metallic strips has also been completed. Composites containing carbon filaments in an epoxy matrix are promising although to date no completely satisfactory strip of this type has been developed. A brief attempt was also made to use homogeneous, non-metallic strips, but to date tests with these have offered no encouragement.

At one time it was thought that fatigue cracks were probably initiated at the edges of the strips and so tumbling and other edge finishing procedures were tried. None of these had any effect on the strip life.

DISCUSSION

The facts uncovered by the investigation can be summarized as follows:

1) The maximum stress in the strip is about 40% higher than that calculated assuming simple bending in the first mode but is still far below the endurance limit of the strip material. See Fig. 9 for effect of valve design on peak stress.

2) Failure of the strips starts with branched cracks on the guard side and these lead to failure by bending fatigue.

3) 410 SS is a good choice of material for the strip although it is not clear which of its properties are important. Other materials which should on paper perform better didn't. There is no evidence that our strip material has inclusions, surface defects on other imperfections that reduce the endurance limit or strip life.

4) The use of brass instead of cast iron guards increases the life of the strips from 8×10^6 cycles to 160×10^6 cycles, but after this failures occur rapidly. Use of a non-metallic cushion on the guard increases strip life to more than 210×10^6 cycles.

5) Analysis of the impact stress shows that a non-metallic cushion would be expected to reduce the stress to a very low level, but that the brass should show very little improvement over cast iron.

6) Reducing the peak stress 30% by reducing the valve lift increases the strip life from 8×10^6 to 70×10^6 cycles, but failures then occur rapidly. Reducing the peak stress 5% by using a sine contour, increases strip life to 60×10^6 cycles.

7) Shot peening increases strip life and endurance limit.

The dramatic effect of using a soft guard and the influence of the guard profile seem to prove that the initial damage to the strip is caused by the impact of the strip on the guard. However, the mechanism by which the impact causes failure is not clear. The maximum stresses that can be calculated or measured in the strip during impact are less than half the strip endurance limit and the branched cracks are not characteristics of normal fatigue failures. The results do agree well with these obtained on an impact rig (2) by Svenzon and the rig results suggest that failure may be initiated away from the actual contact zone between strip and guard. This apparently rules out fretting corrosion or the effects of the guard surface finish as a cause of failure. It seems that stress waves set up during impact must cause the failures. The fact that failures occur when the stresses measured or calculated are very low is explained either by our not having considered the correct stress wave patterns or to the effects of the high strain rate.

The improvement in strip life with reduced stress could be either due to the rate of branched crack formation being dependent on the longitudinal stress at the guard side surface of the strip or more likely to the fact that the branched cracks have to progress further before fatigue failure will be initiated when the stress is lower.

CONCLUSIONS

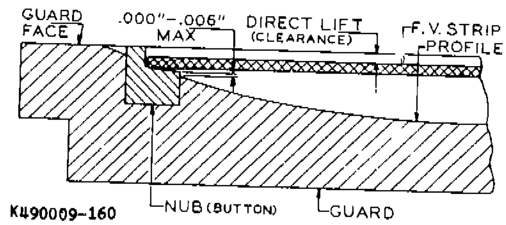
This work has confirmed the suspicions of compressor designers on the effect of impact on valve life and provides some information on ways to design for higher impact velocities. Recently Soedel (1) and Svenson (2) have attempted to investigate this failure mechanism in more detail and it is hoped that the work described here will reinforce the importance of those studies.

As a result of this work we have a new understanding of one of the failure modes of feather valves and have found several design features that can be used to increase the valve lift and hence improve compressor performance without decreasing valve reliability. From this practical point of view the work was very successful. However, we still have little under.tanding of the actual failure mechanism in impact related failures and there is a definite need for further work on this topic.

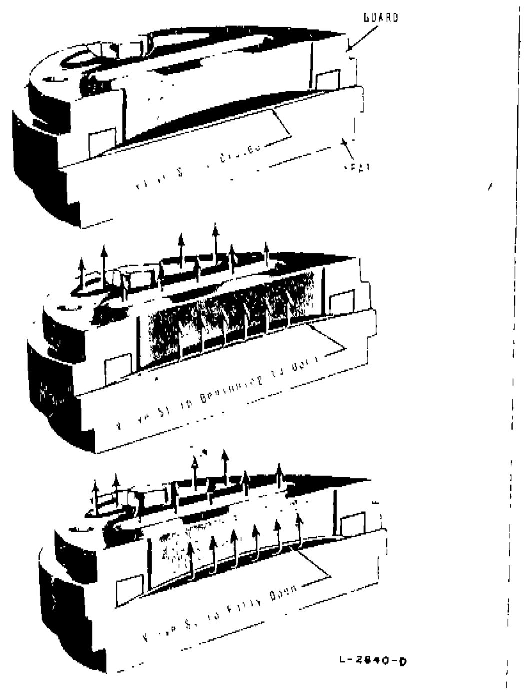
REFERENCES

1. SOEDEL, W., "On Dynamic Stresses in Compressor Valve Reeds or Plates During Colinear Impact on Valve Seats". Proceedings of the 1974 Purdue Compressor Technology Conference, p. 319

2. SVENZON, M., "Impact Fatigue of Valve Steel". Proceedings of the 1976 Purdue Compressor Technology Conference, p. 65



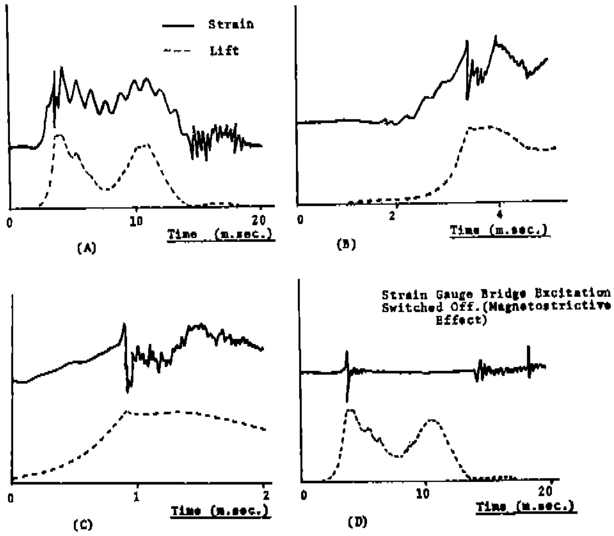
ENLARGED SECTIONAL VIEW THRU GUARD WITH F.V. STRIP IN POSITION



FEATHER VALVE STRIP OPERATION IN VALVE

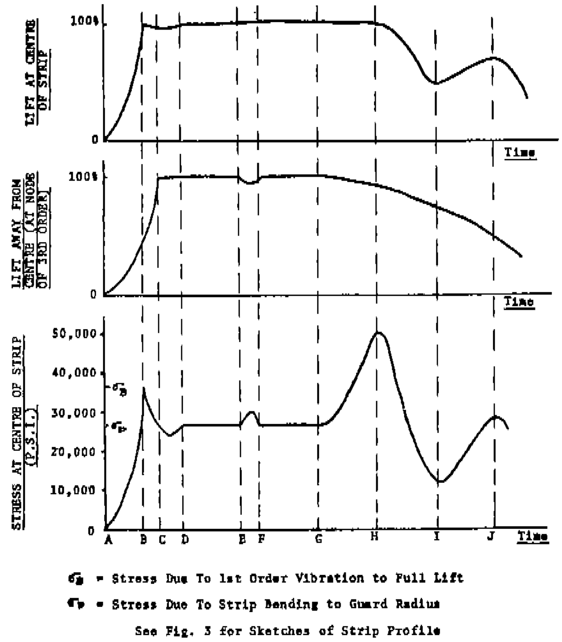
FEATHER VALVE

Fig. 1



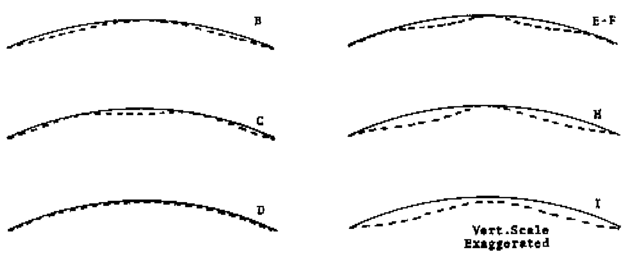
LIFT AND STRAIN GAUGE DIAGRAMS DURING VALVE LIFT AND IMPACT

Fig. 2



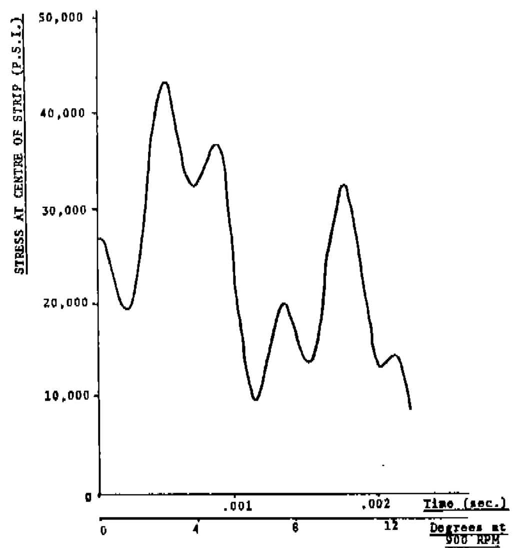
LIFT AND STRAIN DIAGRAMS TO ILLUSTRATE HYPOTHESIS ON INTERACTION OF STRIP AND GUARD

Fig. 4



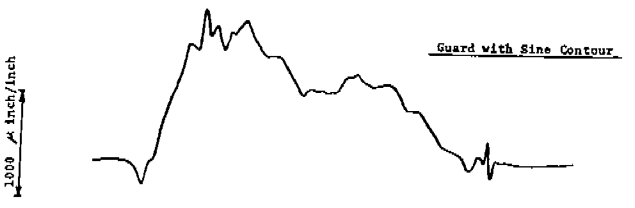
DIAGRAMS TO ILLUSTRATE STRIP PROFILES FOR EVENTS SHOWN IN FIG. 4

Fig. 3



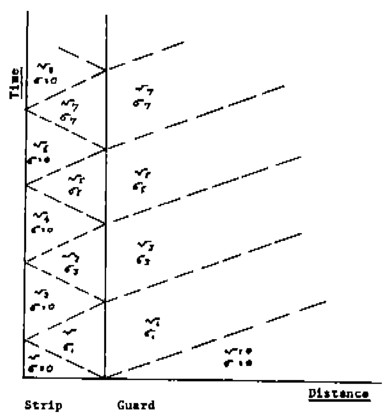
CALCULATED STRESS IN STRIP FOLLOWING RELEASE FROM CIRCULAR PROFILE

Fig. 5



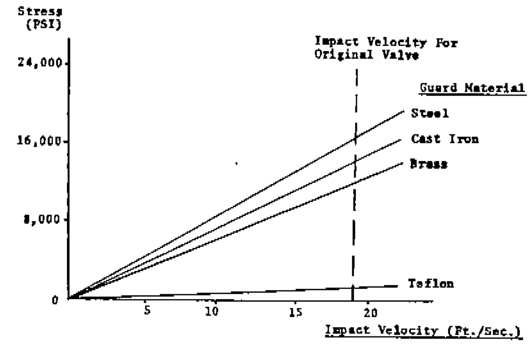
STRAIN GAUGE DIAGRAMS - .032" STRIP THICKNESS

Fig. 6



WAVE DIAGRAM

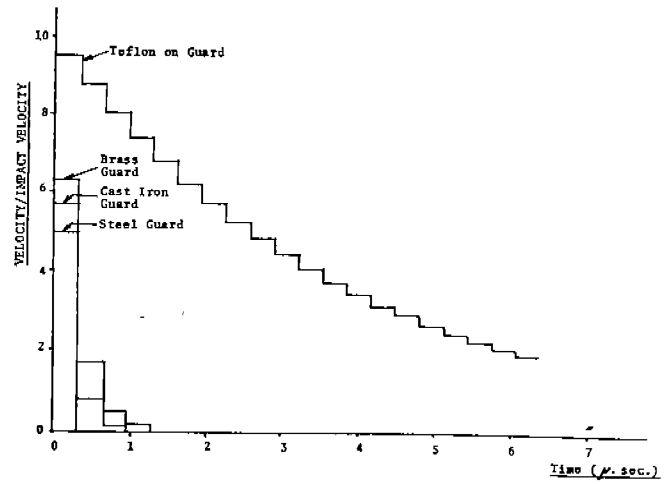
Fig. 7



STRESS CAUSED IN STEEL STRIPS

BY IMPACT ON GUARDS OF DIFFERENT MATERIALS

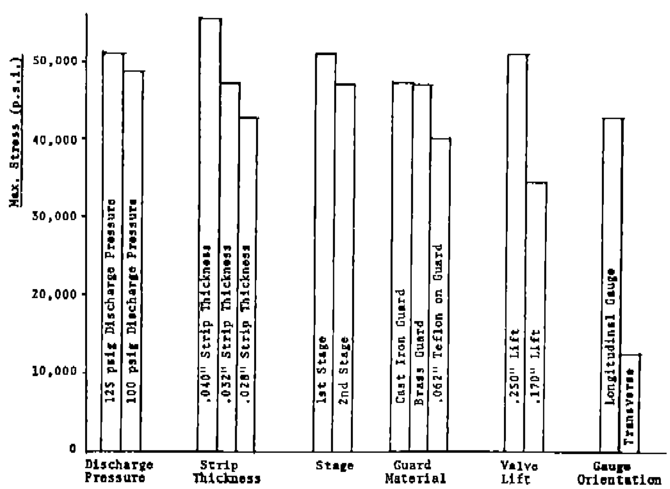
Fig. 8A



VELOCITY OF CONTACT POINT AFTER STRIP

HITS GUARD

Fig. 8B



SUMMARY OF STRAIN GAUGE TEST RESULTS

Fig. 9