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ABSTRACT

An experimental and analytical study concerning the impact fatigue characteristics of discharge flapper valves with different valve thicknesses and valve seat conditions was performed.

In the present experiment, the impact velocity corresponding to the impact fatigue limit was determined from a relationship between the impact velocity and impact fatigue failure. In addition to the impact fatigue measurement, an analysis was made of the impact stress distribution in the valve after contact between the valve and the seat. The computations were carried out using the nonlinear dynamics three dimensional finite 8-node element method. The computed results on the impact fatigue strength were shown to be in good agreement with the corresponding measured data.

INTRODUCTION

In the design of efficient compressor valves from the valve durability view point, a fundamental important problem facing the designers is that of obtaining information about impact fatigue characteristics of the valve.

Since measurements of impact stress on valves which are needed to determine impact fatigue are very difficult to perform in the actual valve and are subject to accuracy limitations, a few attempts have been made to experimentally evaluate impact fatigue with impact velocity or impact intensity(1-2). Unfortunately, it should also be noted that such evaluation methods have the serious disadvantage of being extremely time consuming in case of actual measurements. Further, the majority of the existing studies concerning impact fatigue have been devoted to analytical studies of impact stress (3-8). Most of the analytical works have been done on the elastic wave propagation and traveling in the valves and its component parts (3-7). On the other hand, the impact stress for oblique impact in the valve having an every rounded edge has also been investigated (8). It should be noted that the impact stress studies reported in the existing literature were carried out for a limited range of geometric factors such as valve shape , valve thickness and valve seat shape.

The objective of the present study is to analytically predict the impact stress of flapper valves in the further extended range of the geometrical factors and to determine the impact velocity corresponding to the impact fatigue limit by employing the valve impact fatigue apparatus developed in the present study in order to verify validity of the present analysis. In addition, another feature of the present study is to examine the relationship between the obtained results of the impact stress and the impact fatigue limit in order to propose an evaluating method of the impact fatigue.

EXPERIMENTAL APPARATUS AND EXPERIMENTAL PROCEDURE

Experimental Apparatus

The experiments were performed using a specially designed apparatus simulated the actual valve motion in the compressor in order to evaluate impact fatigue characteristics of valve. The experimental apparatus is schematically shown in Fig. 1. It consists of a 7.5kW compressor to produce pressurized air up to 98MPa, an impact fatigue test stand and pressure control valves. The impact fatigue test stand includes basically the test valve assembly, the pressure vessel and the supply air chamber. The pressure vessel housed the test valve assembly was made up of steel plates with a thickness of 20mm. It was fastened with bolts on the supply air chamber to facilitate a hermetically sealed installation. In addition, a discharge air outlet was drilled in the upper plate of the vessel and was connected to a pressure control valve in the vessel. The test valve assembly to install valve specimens on the discharge air outlet ports was fabricated from a steel block.

To made the impact fatigue testing of the valve specimens, four discharge air outlet ports to load on the valve specimens by compressed air was drilled in the assembly. A supply air passage to provide the discharge air outlet ports with compressed air was also opened in the center of the assembly. Accordingly, an inlet port of the supply air passage was fitted to an outlet port of the supply air chamber.

As seen in the figure, the supply air chamber was designed for producing short-duration compressed air pulses in order to cyclically open and close the valve specimens installed in the valve test assembly. The chamber was comprised of a shaft support and a rotating slotted shaft connected to a variable speed motor. The shaft support was opened both an air port to vent the compressed air to the supply air passage and an air port to exhaust into the laboratory room.

The rotating slotted shaft having both supply air ports and exhaust air ports was mounted in the shaft support with an inside diameter of 40mm and a length of 300mm and also was driven by the variable speed motor. Four supply air ports, P1, and four exhaust air ports, P2, having each rectangular open dimension of 20mm × 5mm were drilled at 90° intervals around the circumference of the rotating slotted shaft, respectively.

In the figure, the solid arrow represents the flow path for the case that the valve specimens were opened by the compressed air vented through the supply air chamber to the valve test assembly. On the other hand, the dotted arrow represents the return flow path for the case that the valve specimens were closed. Consequently, the compressed air pulse was generated by rotation of the rotating slotted shaft.

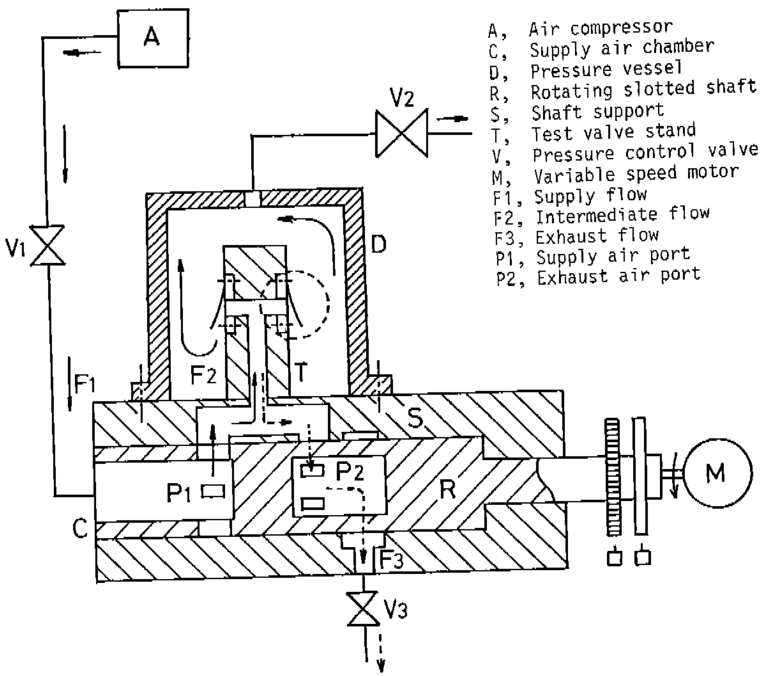


Fig.1 Schematic diagram of valve impact fatigue apparatus

A detail of the installation of the test valve specimens in the test valve assembly is presented in Fig. 2. Each of four valve specimens was installed in each of the four discharge air outlet ports, as illustrated in the figure. Each valve specimen was sandwiched between the valve seat and a backing plate, and was also fastened with a bolt. A central 10mm dia. hole in the backing plate was provided for inserting an eddy current transducer employed to measure the displacement of the valve specimen. The transducer was fixed with a thermosetting resin to the backing plate. An effort was made to align the installed transducer surface with the valve seat in order to achieve smooth movement of the valve specimen.

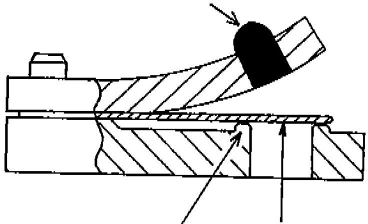


Fig.2 Detail of installation of test valve specimen in test valve assembly

The shape and thickness of the tested valve specimens are presented in Fig. 3. The specimens constructed from stainless steel having a chemical composition listed in Table 1 were blanked from the strip parallel with the rolling direction. Finally, the edge of the specimens were ground and polished. The properties of the tensile strength and fatigue limit in the specimen materials are listed in Table 2.

Furthermore, three different shapes of the valve seat were tested to examine the effect of valve seat shape on the impact fatigue strength of the valve. The shapes and dimensions of the valve seats employed are shown in Fig.4. As shown in the figure, the flat shaped, the half circular shaped and the quarter circular shaped valve seats were selected for use in the present study.

In addition to the instrumentation already mentioned, a piezoelectric pressure transducer was screwed through the air passage wall of the shaft support to measure the compressed air pulse.

Table 1 Chemical composition of test valve specimen (wt%)

C	Si	Mn	Cr	Mo
0.37	0.30	0.35	13.5	1.20

Table 2 Tensile strength and fatigue limit in specimen materials

Tensile strength (MPa)	1800
Tensile fatigue limit (MPa)	610 ± 610

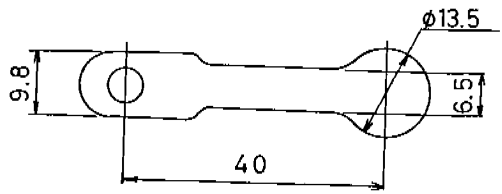
Valve specimen	Thickness(mm)
	0.305
	0.381

Fig.3 Shape and thickness of test valve specimens

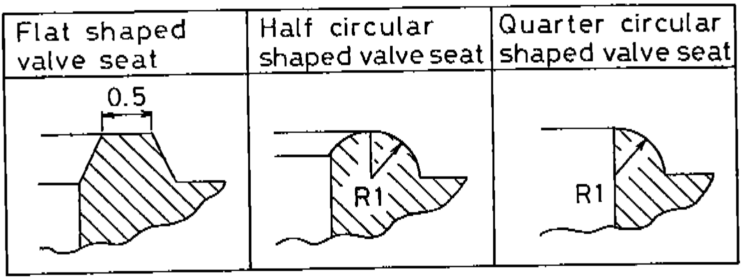


Fig.4 Three different shapes and dimensions of valve seats

Experimental Procedure

In order to make check on reproducibility of the present apparatus simulated the actual compressor, preliminary runs for verifying the compressed air pressure pulse generated in the valve assembly were carried out. In each run, the installed valve specimens were operated by short-duration compressed air pulses which were achieved by regulating the rotational speed of the rotational slotted shaft. The rotational speed was held constant at $25s^{-1}$ (1500rpm). This corresponds to the operating valve specimen cycle of 100Hz.

The impact velocity of the valve specimen defined as the valve velocity immediately before the first contact between the valve specimen and valve seat was varied from 4 to 7m/s. This was done by controlling each of the supply air pressure, the exhaust air pressure and the intermediate air pressure. In the present study, the impact velocity was calculated by differentiating the measured displacement of the valve specimen. The displacement of the valve specimen was sensed by means of the transducer which was calibrated after mounted on the backing plate. The impact fatigue measurements were made at the supply air temperature of approximately 40 °C. The fatigue limits at 10^7 loading cycles were determined according to stair-case method.

STRESS ANALYSIS

In the present study, analysis was performed using a nonlinear dynamic three dimensional finite 8-node elements method to calculate displacement, impact velocity and impact stress of the valves (9). Both analytical models included element meshes of the valve and seat being investigated here are shown in Fig. 5. As seen clearly in the figure, valve-overhang was considered in the analytical model. The shape and dimension of the valve model used was identical to those of the actual valve. Also the shapes of the seat model corresponded to those employed in the present experiments. The finite element mesh was deployed so that the points were finely spaced adjacent to the contact zone. The minimum mesh size was $0.482mm \times 1.00mm \times t/2$.

In the present analysis, numerical computations were made within a half domain as indicated in Fig. 5 due to geometrical symmetry in the longitudinal direction. Referring to the numerical integration presented in the analytical method, the central difference method was employed. A sufficiently small time increment of 30ns was selected in the present numerical computations.

The displacements and the velocities at the nodes labeled 1-4 and the stresses in the element 1-3 were calculated.

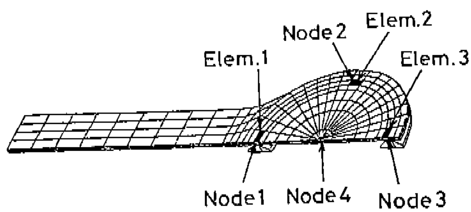


Fig.5 Finite element mesh for a half of valve and seat

RESULTS AND DISCUSSION

Effect of Valve Thickness on Impact Fatigue Characteristics

Fig. 6(a) and 6(b) show both calculated results of velocity and stress for two different thicknesses as a function of time, respectively. In the figures, three velocities and three stresses were calculated at node 1 - node 3 and in element 1 - element 3, respectively. In addition, the above calculated results were for the velocity at node 4 of approximately 5.5 m/s. Inspection of both figure results reveals that the largest impact velocity is obtained at the node 3. In regards to the stress results, it is apparent that both the impact stresses at node 3 increase rapidly after the valve touched the seat, and also reach the maximum of approximately 590 MPa. Further, it can be understood that the difference between the contact time at node 1 and that at node 3 for t=0.381mm becomes smaller compared with that at node 3 for t=0.305mm. This implies that the valve for t=0.381mm having a larger stiffness closes faster than that for t=0.305mm.

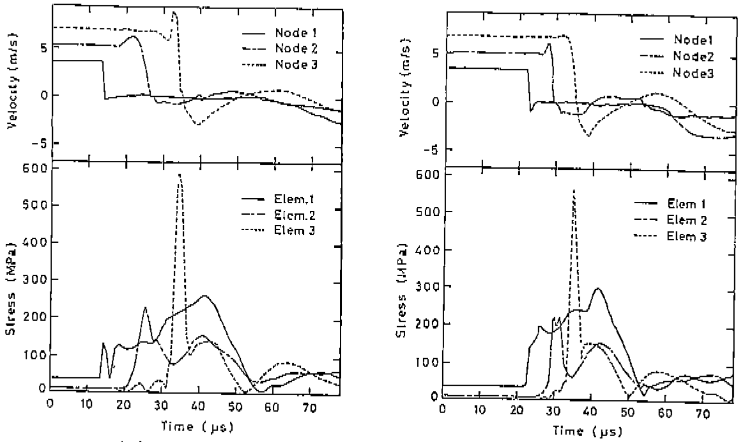


Fig.6 Calculated results of velocity and stress for two different valve thicknessesas a function of time

Fig.7 shows the variations of the impact stress in element 3 with the impact velocity at node 4 for two different thicknesses of the valve. In the figure, the impact velocity at node 4 corresponds to available data measured in the experiments. The impact stress in element 3 appearing in the figure represents a von Mises stress in element 3 in which the maximum stress occurs after the valve touches the seat. Also, the solid and dashed line in the figure correspond respectively to t=0.305mm and t=0.381. From this figure, it is seen that for both cases, the impact stress increases linearly with an increase in impact velocity. Moreover, although the valve thickness appears as a parameter, it has only a slight influence. This finding suggests that the impact fatigue strength for t=0.305mm is almost the same as that for t=0.381mm.

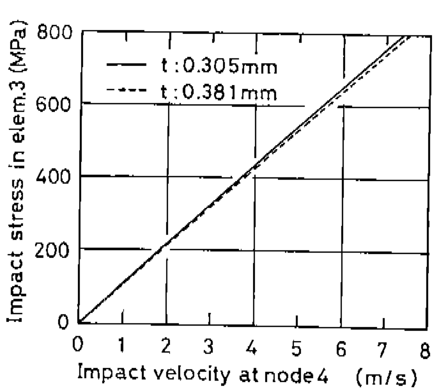


Fig.7 Variation of impact stress in element3 with impact velocity at node4 for two different valve thicknesses

Table 3 presents the experimental data of the impact velocity corresponding to the impact fatigue limit for both t=0.305mm and t=0.381mm. It is found from the present experiments that a significant difference in the impact fatigue strength between for t=0.305mm and for t=0.381mm does not appear. The calculated impact stress corresponding to the impact velocity to the impact fatigue limit was found to be 590 MPa, and also yielded to be approximately equivalent to the tensile fatigue limit of 610 MPa. A comparison of the measured result with the calculated result reveals that the present analysis is available for determining the impact stress.

Table 3 Experimental data of impact velocity corresponding to impact fatigue limit for two different valve thicknesses

Valve thickness t mm	Impact fatigue limit m/s
0.305	5.6
0.381	5.5

As was shown earlier in Fig. 6, it should be noted that although both the impact velocities at node 3 for t=0.305mm and for t=0.381mm were markedly different from both the impact velocities at node 4 for t=0.305mm and for t=0.381mm were held the same value, and also the impact stress in element 3 for t=0.305mm agreed with that for t=0.381mm.

In order to discuss the above mentioned finding, mention must also be made here of comparisons with the impact velocity results at node 3 and those at node 4 as shown in Fig. 8(a). In addition to the impact velocity results, the relationship between the impact stress in element 3 and the impact velocity at node 3 was demonstrated in Fig. 8(b). From an examination of Fig. 8(a), it is seen that for a thinner valve the impact velocity at node 3 becomes larger. This takes place because for a thinner valve the whip-lash effect becomes more pronounced. With regard to effect of the valve thickness on the impact stress, the impact stress in element 3 of Fig. 8(b) for a thinner valve exhibits to become smaller. This can be easily explained from the wave propagation theory proposed by Dusil,et al. (1).

From both Fig.8(a) and Fig.8(b) it is deduced that for the same impact velocity at node 4, the impact stress in element 3 for t=0.305mm is equal to that for t=0.381mm.

On the other hand, the calculated velocity at the very tip was found to attain a value of 17 m/s. It is considered that such an increased velocity at the very tip is achieved due to the whip-lash effect. Consequently, this finding suggests that the impact stresses calculated from the present analysis with the over-hang effect become smaller compared with those having no over-hang effect.

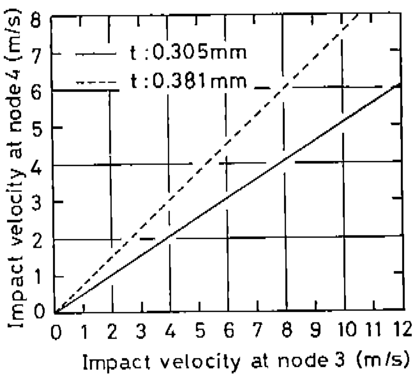


Fig.8(a) Comparison with impact velocity at node4 and that at node3

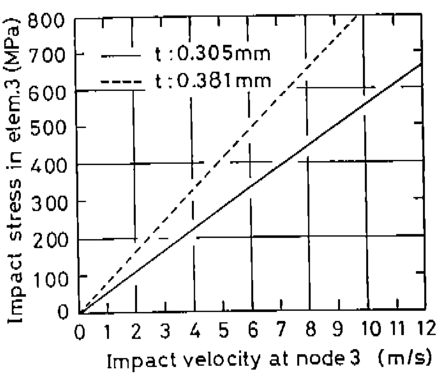


Fig.8(b) Relationship between impact stress in element 3 and impact velocity at node 3

Fig. 9 shows the effect of the valve thickness on predicted results of the impact stress ranging from t=0.2mm to t=0.5mm for impact velocity of 5.5 m/s. In the figure, the stress ratio represents the ratio of the impact stress for a given thickness to that for a thickness of 0.305mm. The valve thickness is not quite sensitive to the impact fatigue strength.

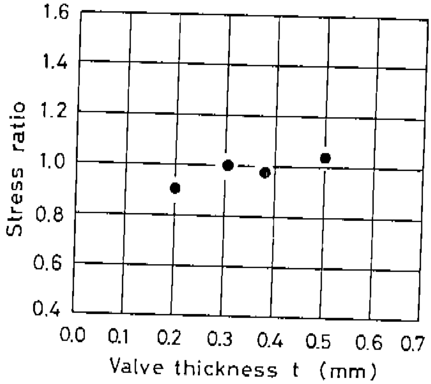


Fig.9 Effect of valve thickness on impact stress for impact velocity of 5.5m/s

Effect of Valve Seat Geometry on Impact Fatigue Characteristics

Fig. 10 shows the effect of the valve seat with three different geometries on the impact stress. As seen in the figure, it is evident that for the case of the flat seat the lowest impact stress occurs, but the largest measured data of impact velocity is presented in Table 4.

Further, each of the impact velocities evaluated using the obtained impact stress value of 590 MPa are 3.6 m/s for the quarter circular shaped seats, 4.2 m/s for the half circular shaped seat, 5.6 m/s for the flat shaped seat, respectively. It is seen that the present analysis of the impact velocity for the different seat geometries predicts the measured data within 13%.

Fig. 11 shows the variation of the impact stress with the seat width/contact area for the flat shaped seat. The stress ratio appearing in the figure represents the ratio of the impact stress at certain valve width to that at seat width of 0.5mm. It is seen from the figure that the larger seat area, the smaller is the impact stress, namely the greater is the impact fatigue strength. A good agreement with the present results and the measured data by Dusil, et al. (1) suggests that the present analysis is available for evaluating the

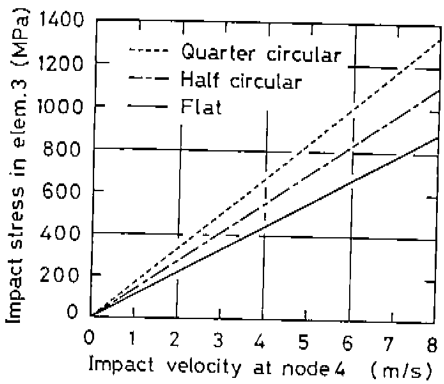


Fig.10 Effect of valve seat with three different geometries on impact stress

Table 4 Experimental data of impact velocity to impact fatigue limit for three different valve seat shapes

Valve seat	Impact fatigue limit m/s
Flat	5.6
Half circular	3.8
Quarter circular	3.2

the impact stress of the valve for valve seats with different geometries.

Based on the obtained information, a technique to provide a reasonable guide for designing suitable valves was discussed below: First, the impact velocity corresponding to the impact fatigue limit is determined from the calculated relationship between the impact velocity and impact stress obtained from a wide range of design parameters. On the other hand, impact velocity is also measured in the actual case. Finally, by comparing the calculated impact velocity with the measured one, the safety factor is evaluated so that a suitable valve can be selected.

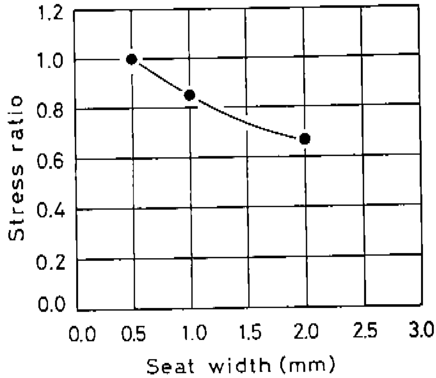


Fig.11 Variations of impact stress with seat width for flat shaped seat

CONCLUSIONS

The impact stress and impact velocity of the compressor flapper valves have been investigated here both experimentally and analytically. In the analytical/numerical work, the impact velocities at node 1-4 and impact stresses in element 1-3 were solved by employing the nonlinear dynamics three dimensional finite element method. Measurements were performed to obtain the impact velocities of the valves for two different thicknesses and to three different valve seats.

The obtained impact stress results were found to be uninfluenced by the variations of valve thickness. The smallest impact stress was obtained in case that a flat shaped seat was used. The calculated impact velocities which correspond to the impact fatigue limit were found to be in a good agreement with the measured data.

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