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A Fundamental Study on Valve Impact Stress in Refrigeration Compressor

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A FUNDAMENTAL STUDY ON VALVE IMPACT STRESS IN REFRIGERATION COMPRESSOR

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ABSTRACT

The purpose of this study is to get fundamental knowledge of the valve impact stress by clarifying wave propagation mechanism in two bars which are simulated to a valve seat or valve stop and a valve. Axially uniform stress in finite cylindrical elastic bars subjected to velocity-impact loading is considered on the bases of elementary equations of energy. A technique is developed for measuring the strains transmitted by the longitudinal impact of two bars and the variations in the strain are measured under three kinds of bar lengths and two different restraints at the end point. Comparing the theoretical results with the experimental results, the effects of the bar lengths and restraint conditions on the propagation of the stress wave along a slender cylindrical bars are examined. On the bases of these results, the impact duration and maximum impact stress of valves and the transient response of strain gauges are discussed. Moreover, predominant impact stress affecting impact fatigue failure of valves is also discussed.

INTRODUCTION

A valve of a refrigeration compressor is subjected to two types of fatigue damages; bending fatigue and impact fatigue. The bending fatigue implies manageable problems because the motion of the valve under compressor operating conditions would be estimated in blueprint stage of the compressor design by using the simulation programs of valve dynamics, and the fatigue strength of valve materials would be evaluated by using a commercially available fatigue testing machine. On the contrary, the impact fatigue would be caused by repeated impact against a valve seat and valve stop, being characterized by tearing off of chips from edges. The phenomena have complex nature of both the shock effects and the engineering structures. The problem of the compressor valve, therefore, is of more considerable importance for withstanding high impact shock.

The dispersive nature of an elastic wave traveling in a compressor component have already been investigated by Soedel[1],[2], Pendeya[2] and Simonitsch[3]. Impact fatigue testings have been conducted by Svenzon[4] and Dusil[5]. Davis[6] has also shown design procedure giving the impact velocities of the valves. Although considerations such as these studies have provided the motivation for extensive studies of valve impact stress, they could not quantitatively evaluate the apparent vital role of the complex phenomena. In order to clarify complex nature of valve impact failure, therefore, further investigations seem to be needed in conjunction with the stress that occurs due to the compressor operating loads.

Accordingly, this study is carried out in order to get fundamental knowledge of the valve impact stress by clarifying the wave propagation mechanism and by developing the measurement technique of the impact strain. In this paper, a theoretical solution of an elementary theory for wave propagation is obtained with velocity-impact loadings at the impact end of two finite elastic bars. A technique is developed for measuring the strains transmitted by the longitudinal impact of two bars which are simulated to the valve seat or valve stop and the valve. Comparing the theoretical results with the experimental results, the effects of bar lengths and restraint conditions on the propagation of the stress wave along a slender cylindrical bar are examined. On the bases of these results, the impact duration and maximum impact stress of valves and the transient response of strain gauges are discussed. Moreover, predominant impact stress affecting impact fatigue failure of valves is also discussed.

THEORY

Equations of Impact Stress

In this study, an elementary theory of the wave propagation mechanism was examined to get more accurate information on the valve impact stress. The problem of impact stress transmitted inside a rod have already been investigated by Ripperger (7), Skalak (8), Matsumoto (9), and so on. On the bases of these studies, the impact wave propagation mechanism was analysed in this study.

Figs. 1 (a) and (b) show two bars with different cross sectional areas of A_1 and A_2 before and after striking each other longitudinally with velocities of V_1 and V_2. In this paper, the left and right bars shown in Fig. 1 are called striking and struck bars and are distinguished with suffixes 1 and 2, respectively. Denoting the cross sectional area, length and weight per unit volume of the striking and struck bars by A, ℓ and γ, total kinetic energy KE_1 in two bars before striking is expressed by

$$KE_1 = \frac{1}{2} A_1 \ell_1 \frac{\gamma_1}{g} v_1^2 + \frac{1}{2} A_2 \ell_2 \frac{\gamma_2}{g} v_2^2 \tag{1}$$

where g is gravitational acceleration. After striking, the compression waves σ_1 and σ_2 start to travel along both bars with velocities of c_1 and c_2, expressed by

$$c_1 = \sqrt{\frac{E_1 g}{\gamma_1}} \tag{2}$$

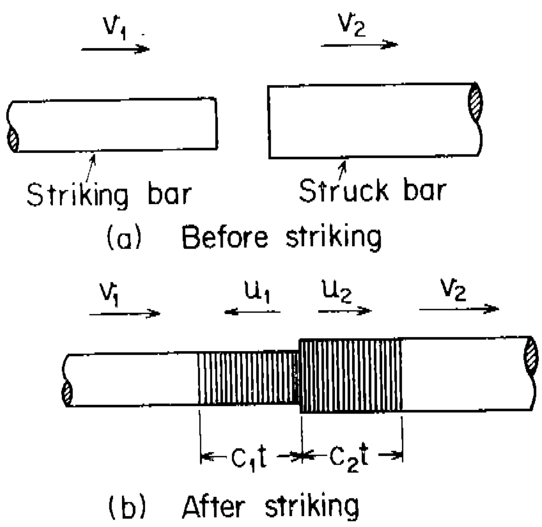


Fig.1 Schematic diagram of longitudinal impact of two bars.

$$c_2 = \sqrt{\frac{E_2 g}{\gamma_2}} \tag{3}$$

where E is modulus of elasticity.

At definite instant t after striking, the compression waves σ_1 and σ_2 travel within the distance c_1t and c_2t from the plane in contact between two bars. Assuming that the velocities of the particles within the regions of c_1t and c_2t are u_1 and u_2, the kinetic energy KE_2 and the strain energy SE_2 in two bars in these regions are expressed by

$$KE_2 = \frac{1}{2} A_1 c_1 t \frac{\gamma_1}{g}(v_1 - u_1)^2 + \frac{1}{2} A_2 c_2 t \frac{\gamma_2}{g}(v_2 + u_2)^2 \tag{4}$$

$$SE_2 = \frac{1}{2} A_1 c_1 t \frac{\sigma_1^2}{E_1} + \frac{1}{2} A_2 c_2 t \frac{\sigma_2^2}{E_2} \tag{5}$$

In other regions of two bars, only kinetic energy KE_3 remains, expressed by

$$KE_3 = \frac{1}{2} A_1 (\ell_1 - c_1 t)\frac{\gamma_1}{g} v_1^2 + \frac{1}{2} A_2 (\ell_2 - c_2 t)\frac{\gamma_2}{g} v_2^2 \tag{6}$$

Assuming that there is no energy loss during impact, and that the velocities of two bars at the plane in contact are equal, the following equations are obtained.

$$KE_1 = KE_2 + KE_3 + SE_2 \tag{7}$$

$$v_1 - u_1 = v_2 + u_2 \tag{8}$$

From Eqs. (1)∿(8), the impact stresses σ_1 and σ_2 of two bars are expressed by

$$\sigma_1 = \frac{A_2\sqrt{E_2\gamma_2}}{A_1\sqrt{E_1\gamma_1} + A_2\sqrt{E_2\gamma_2}} \times \sqrt{\frac{E_1\gamma_1}{g}}\,(v_1 - v_2) \tag{9}$$

$$\sigma_2 = \frac{A_1\sqrt{E_1\gamma_1}}{A_1\sqrt{E_1\gamma_1} + A_2\sqrt{E_2\gamma_2}} \times \sqrt{\frac{E_2\gamma_2}{g}}\,(v_1 - v_2) \tag{10}$$

Considering the special case that the struck bar is standstill, and that the cross sectional area and material of two bars are same; $A_1 = A_2$, $E_1 = E_2$, $\gamma_1 = \gamma_2$ and $v_2 = 0$, the following equation is obtained from Eqs. (9) and (10).

$$\sigma_1 = \sigma_2 = \frac{1}{2}\sqrt{\frac{E_1\gamma_1}{g}}\, v_1 \tag{11}$$

An examination of Eqs. (9)∿(11) shows that the impact stress depends only on the weight per unit volume, modulus of elasticity, and the cross sectional area of the bar.

Reflected and Transmitted Stresses at Discontinuity in Cross Section

The preliminary procedure described above is now applied to a bar containing a discontinuity in cross section. Fig. 2 shows the reflected and transmitted stresses at a discontinuity in cross section. As can be seen in Fig. 2, the reflected and transmitted stresses are denoted by $\alpha\sigma$ and $\beta\sigma$, respectively. When the impact stress propagates to a discontinuity in cross section, strain energy SE_3 in the region of c_1t is expressed by

$$SE_3 = \frac{1}{2} A_1 c_1 t \frac{\sigma^2}{E_1} \tag{12}$$

At definite instant t after the stress wave reaches the discontinuity, the total strain energy SE_4 produced by the reflected and transmitted stresses is expressed by

$$SE_4 = \frac{1}{2} A_1 c_1 t \frac{(\alpha\sigma)^2}{E_1} + \frac{1}{2} A_2 c_2 t \frac{(\beta\sigma)^2}{E_2} \tag{13}$$

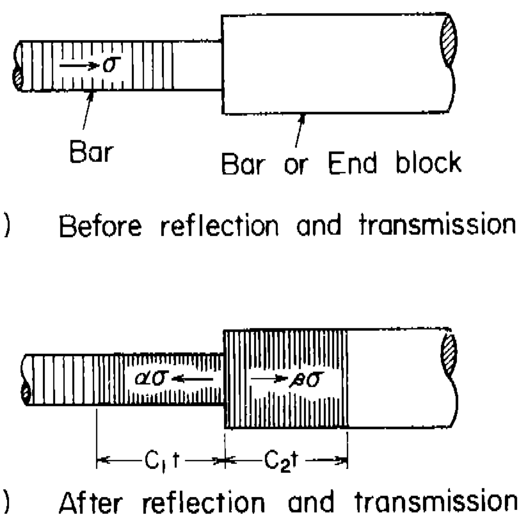


Fig.2 Reflected and transmitted stresses at discontinuity in cross section.

Replacing $SE_3 = SE_4$, the following equation between α and β is obtained.

$$1 - \alpha^2 = \frac{A_2\gamma_1 c_1}{A_1\gamma_2 c_2}\beta^2 \tag{14}$$

Considering that the velocities of both members at the plane in contact are equal, the following equation is also obtained.

$$1 - \alpha = \frac{c_1\gamma_1}{c_2\gamma_2}\beta \tag{15}$$

From Eqs. (14) and (15), α and β are expressed by

$$\alpha = \frac{A_2\sqrt{E_2\gamma_2} - A_1\sqrt{E_1\gamma_1}}{A_1\sqrt{E_1\gamma_1} + A_2\sqrt{E_2\gamma_2}} \tag{16}$$

$$\beta = \frac{2A_1\sqrt{E_2\gamma_2}}{A_1\sqrt{E_1\gamma_1} + A_2\sqrt{E_2\gamma_2}} \tag{17}$$

Since E_2 is equal to zero for a free boundary, $\alpha = -1$ and $\beta = 0$ are obtained from Eqs. (16) and (17). This means that at a free end the impact stress reflects with equal amplitude, but opposite in phase. In the case of a fixed end, $\alpha = 1$ and $\beta = 0$ are obtained from Eqs. (16) and (17) because A_2 is infinite. This means that at a fixed end the impact stress reflects with equal amplitude, and same in phase. Eqs. (16) and (17) are also applied to a bar composed of two materials fused together at some point along its length.

TEST EQUIPMENT AND TEST PROCEDURE

Test Equipment

Fig. 3 shows a schematic arrangement of instrumentation of impact strain. Figs. 4 (a) and (b) are views of the test

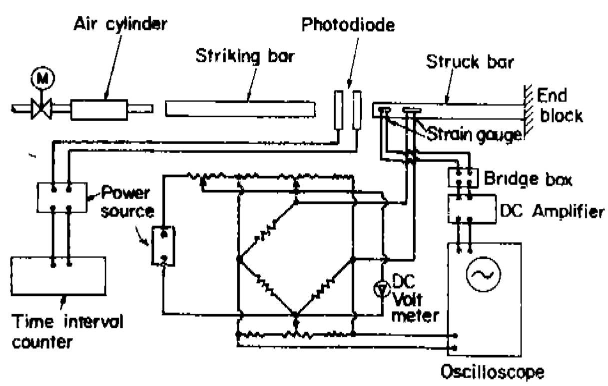
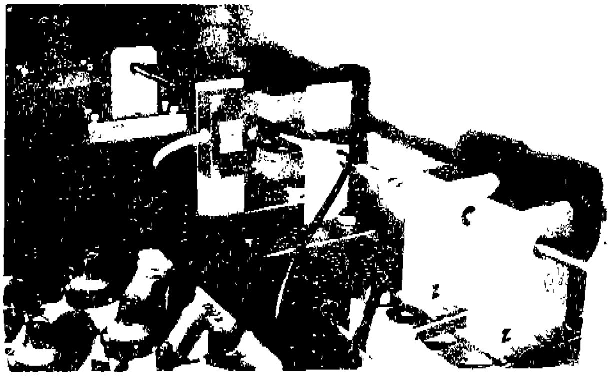
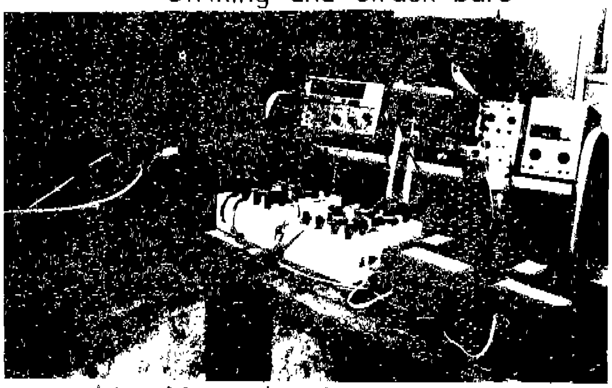


Fig.3 Schematic arrangement of instrumentation of impact strain.

apparatus. Two cold rolled bars, 10 mm in diameter, were used for the test. One was used as a striking bar, the other as a struck bar. Two bars were suspended by needle bearings in the direction parallel to the length. An air cylinder was used to accelerate the striking bar.



(a) Striking and struck bars



(b) Measuring instruments

Fig.4 Views of test apparatus.

Test Procedure

Fig. 5 shows an illustration of locations of strain gauges. Four strain gauges with a gauge length of 2 mm were cemented circumferentially with an angle of 90 degrees at a distance of 140 mm apart from the end of the struck bar. These gauges were connected in series in a bridge circuit shown in Fig. 3, so as to eliminate the effect of flexural strains and to gain increased sensitivity. A pair of dummy gauges were mounted in the bridge box into which the leads from four active gauges on the bar could be plugged to complete the bridge circuit. This arrangement made it possible to more quickly from one pair of gauges to the next with a minimum disturbance in the measuring circuit. The strain gauges were connected in series across a constant current source and oscilloscope. The use of constant-current concept for dynamic strain measurements is a simple method and produces linear data with high sensitivity in a manner which is free from noise generated by lead-wires and contact resistance and their variations with time and temperature. In addition, it permits common grounding of all components for minimum active noise levels in the system.

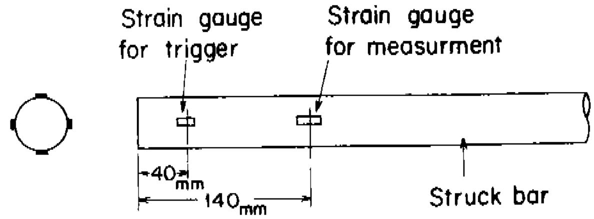


Fig.5 Illustration of locations of strain gauges.

The velocity of the striking bar just before striking was measured by an arrangement consisting of two sets of light sources and photodiodes, 25 mm aparted each other. The time taken by the striking bar cut the two narrow beams of light was recorded by a time-interval counter. With this arrangement, the error in velocity measurements is estimated to be less than one percent.

The recording of the strain history was made by photographing the amplified strain gauge output pattern which appeared during single triggered sweep of the oscilloscope. It is convenient to synchronize the sweep with the impact stress produced in the struck bar. This was accomplished with the signal of the strain gauge bonded at the location apart from the plane in contact with a distance of 40 mm.

Test Conditions

Two different boundary conditions were chosen in this test. One was a free end on the opposite side to the plane struck by the striking bar. This boundary condition was accomplished by inserting a rubber pad with a thickness of 5 mm between the far end surface of the struck bar and an end block. The other was a fixed end. This was accomplished by pressing the struck bar against the end block with a rubber cord.

Velocities ranging from 0.5 m/s to 5 m/s were chosen in this test in taking the valve impact velocity against the valve seat or valve stop into consideration.

Since the impact stress travels inside the finite striking and struck bars, the time interval of the impact stress passing through the location bonding the strain gauge would be affected by the lengths of the striking and struck bars. In this test, therefore, the struck bar with a length of 420 mm was used, and the striking bars with the lengths of 1000, 420 and 280 mm.

Static Test for Calibration of Stress

A stored-energy testing machine was designed and constructed for calibration of the stress by the static loading. With this machine, essentially dead-load testing is achieved. Also, this machine provides close control of the parallelism of the plane in contact and the perpendicularity of the loading axis to the loading faces.

Although the static tests were performed at relatively low strain rates, it was necessary to use dynamic measuring instrumentation to record the rapidly changing stresses which were encountered in this type of test. Strain rates as high as 1 s^{-1} are encountered in dead-load testing as compared to strain rates around 10^{-3} s^{-1} for normal static testing.

THEORETICAL AND EXPERIMENTAL RESULTS

Figs. 6 $\sim$ 9 show typical examples of the stresses developed at the gauge points by an impact velocity of about 3 m/s. The variations in the impact stresses obtained from the elementary theory are also shown in these figures.

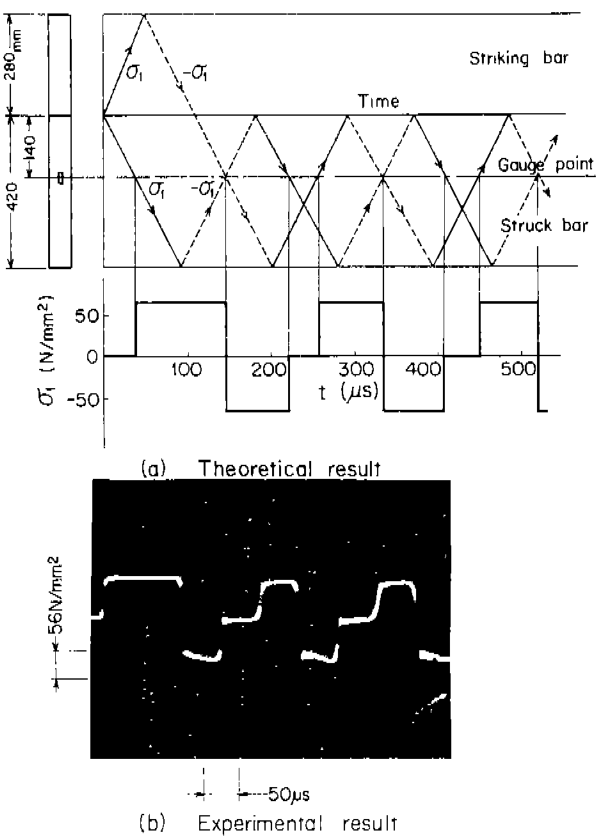


Fig.6 Stress pulse in struck bar (free-free) at $v_1 = 3.3$ m/s.

Free-Free Bar: Figs. 6 (a) and (b) show the theoretical and experimental results obtained with a pair of bar lengths of ℓ_1 = 280 mm and ℓ_2 = 420 mm. It is evident from Fig. 6 (a) that after the bars have struck, a wave of compression proceeds at a velocity of 5.12 x 10^6 mm/s, down the struck bar to the far end, after which a reflected wave of the opposite sign returns relieving the compression. Then, shortly after collision a build-up of compressive stress is recorded by strip shown in Fig. 6 (b). Since the wave form is one of pressure building up to a practically steady value, stress in the region through which the wave front has passed will maintain a constant level until the return of the reflected wave. When the reflect wave in the striking bar reaches the end of the striking bar, where two bars are in contact, this wave transmits to the struck bar. Since at this moment this reflect wave has reduced the pressure between two bars to zero, the bars separate each other and the impact is over. After that, two stress waves in the struck bar travel up and down, reflecting at each end with change of sign. This phenomenon produces such complex variation in stress as can be seen in Fig. 6.

Fig. 7 shows the results obtained with a pair of bar lengths: ℓ_1 = 1000 mm and ℓ_2 = 420 mm. It is clear from Fig. 7 that the

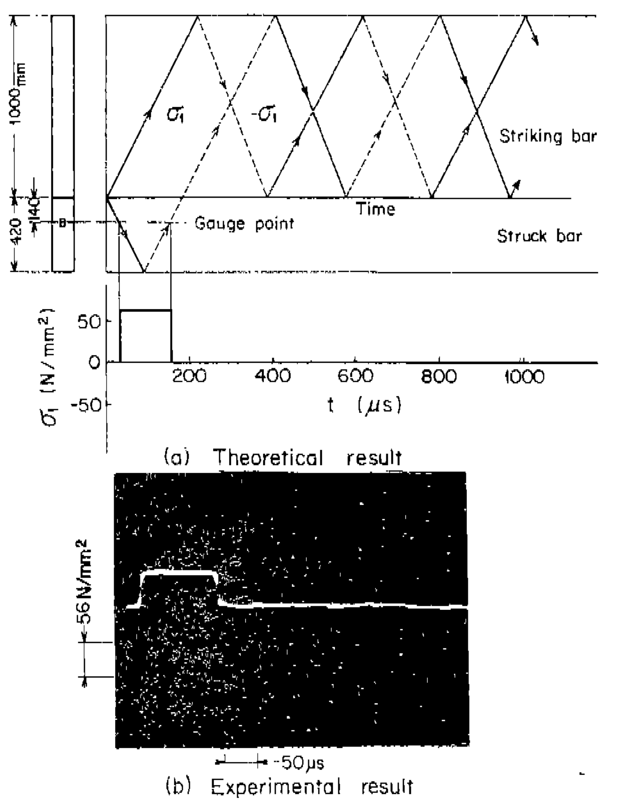


Fig.7 Stress pulse in struck bar (free-free) at $v_1 = 2.9$ m/s.

same evidence is observed, and that the result obtained by the preliminary theory is in good agreement with the test result.

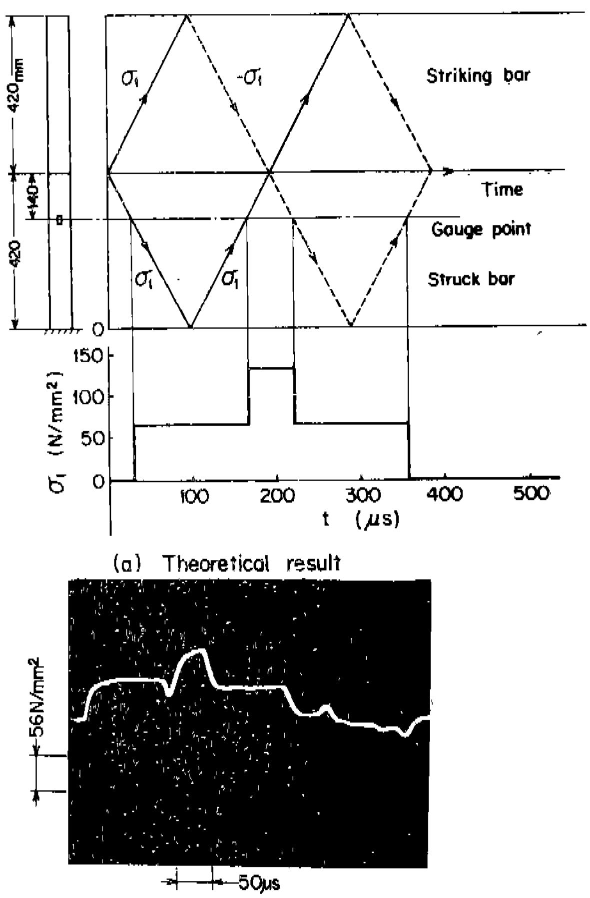
Bar Restrained at One End: Figs. 8 and 9 show the results obtained by two pairs of bar lengths: $\ell_1 = \ell_2 = 420$ mm, $\ell_1 = 1000$ mm and $\ell_2 = 420$ mm, respectively. These results were obtained under the restrained condition at the far end of the struck bar. Since the restrained end of the struck bar was accomplished by pressing the struck bar to the end block with rubber cord, this arrangement could not give complete fixity of the end in the mathematical sense. However, the test results reveal that it provides considerable restraint. In this restrained condition, the results obtained by the preliminary theory are in good agreement with the test results.

DISCUSSION

Impact Duration of Valve

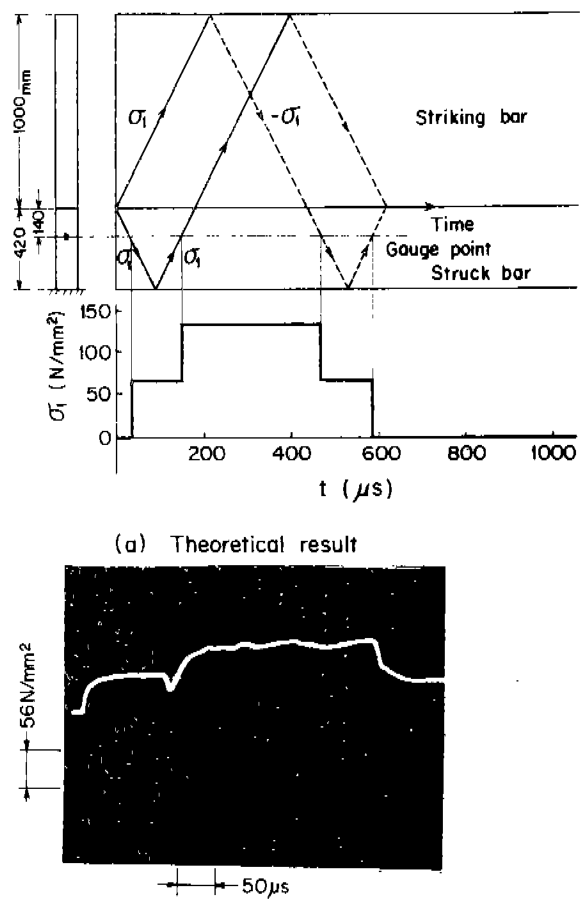
The above preliminary theory and test results demonstrate that the time of each event on the test results is found to agree with the times predicted by the preliminary theory, and that the impact duration can be estimated with high accuracy.

The valve thickness in the range from 0.15 to 2 mm is widely used for refrigeration compressors, especially less than 0.5 mm for high speed compressors. These values are enormously less than the thickness of the valve stop and valve seat. Since this corresponds to the case of $\ell_1 \ll \ell_2$ in this study, it leads to the conclusion that the impact duration is affected by only the material and thickness of the valve, if gray cast iron or steel is used as materials for valve stop and valve seat. Dividing the valve thickness by the velocity of stress wave, therefore, an impact duration within the range from 0.59×10^{-7} to 7.8×10^{-7} s is obtained. After that, the valve will separate from the valve stop or valve seat. This phenomenon may be a cause of the valve chattering, leading to unavoidable problems because of ruling with impact stress propagation.



(a) Theoretical result
(b) Experimental result

Fig.8 Stress pulse in struck bar (free-fixed) at $v_1 = 3.2$ m/s.



(a) Theoretical result
(b) Experimental result

Fig.9 Stress pulse in struck bar (free-fixed) at $v_1 = 3.0$ m/s.

Transient Response of Strain Gauge

In order to examine the transient response of the strain gauge, the rise time was measured in the test. In this test, the time scale on the oscilloscope was exaggerated by 10 times that in wave propagation measurement. As a result, a rise time of about 4×10^{-6} s was obtained. This rise time is much larger than a theoretical value of 0.39×10^{-6} s obtained by dividing the strain gauge length of 2 mm by the velocity of the stress wave propagation.

Four dominant reasons are considered as the causes of this difference. The first is the thickness and materials of the gauge base and the gauge cement. The second is the capacitance between the gauge wire and the struck bar. The third is the parallelism of the plane in contact and the perpendicularity of the loading axis to the loading faces. The fourth is rough of the plane surface in contact. Although further investigations are considered to be needed for clarifying the effect of these factors on the transient response of the strain gauge, the first one becomes more important factor in measuring the impact wave propagation traveling valves with thinner thickness.

As mentioned above, the impact duration is within the range from 0.59×10^{-7} to 7.8×10^{-7} s. Although it is expected to improve the rise time by using strain gauges with shorter length, it can not keep the stress in the region through which the wave front has passed a constant level until the return of the reflected wave. This means that the technique using strain gauges is not so successful to measure the impact stress propagation of the compressor valve.

Maximum Impact Stress of Valve

Fig. 10 shows the maximum impact stress for two restrained conditions; free and fixed restraints at the far end. The test results are also shown in this figure with white circle and cross mark. It is clear from Fig. 10 that the results obtained from the preliminary theory is in good agreement with the test results.

Since the thickness of valves for refrigeration compressors is enormously less than the thickness of the valve stop and valve seat, it corresponds to the case of $A_2 = \infty$ and $v_2 = 0$ in Eq. (9). The maximum impact stress σ_{max} of the valve, therefore, is expressed by

$$\sigma_{max} = \sqrt{\frac{E\gamma}{g}}\, v \qquad (18)$$

The maximum impact stress of the valve is two times the value of σ_1, expressed by Eq. (11) and is equal to the maximum impact stress under restrained conditions.

Authors[(10)~(14)] have both theoretically and experimentally revealed the impact velocity of the compressor valve against the valve seat or valve stop over a wide range of the compressor operating conditions. Fig. 11 shows the relationship between the impact velocity of a suction

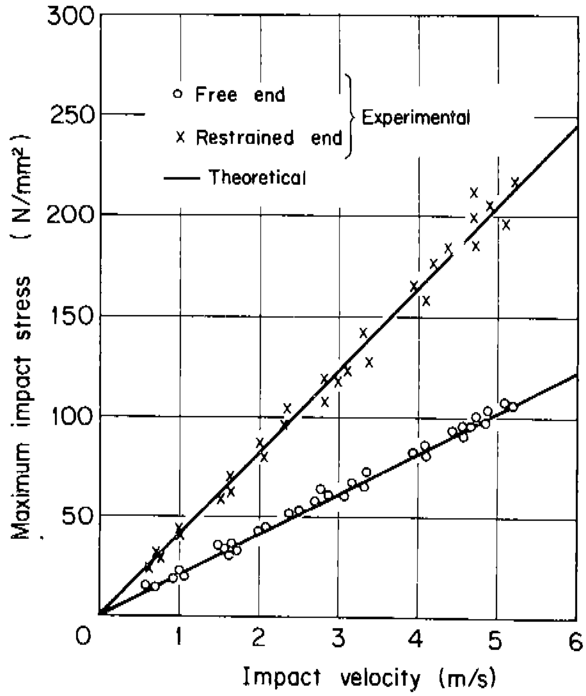


Fig. 10 Maximum impact stresses under free and restrained conditions.

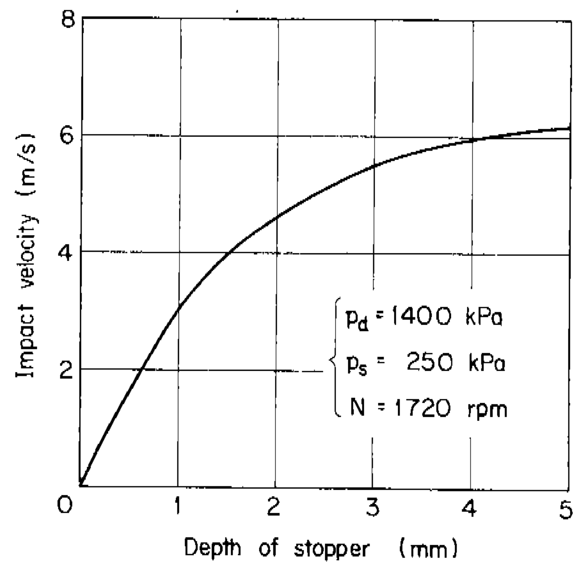


Fig.11 Relationship between impact velocity of suction reed valve and depth of stopper.

reed valve against a stopper and the depth of stopper(13). Fig. 12 shows the relationship between the impact velocity of a discharge ring valve against a stopper and the hight of stopper(10). As can be seen in Figs. 11 and 12, the refrigeration compressors including high speed ones are designed below an impact velocity of about 5 m/s. Svenzon(4) and Dusil(5) have conducted impact fatigue testings and have mentioned that the impact fatigue fracture is created beyond an impact velocity of 6 ∿ 9 m/s.

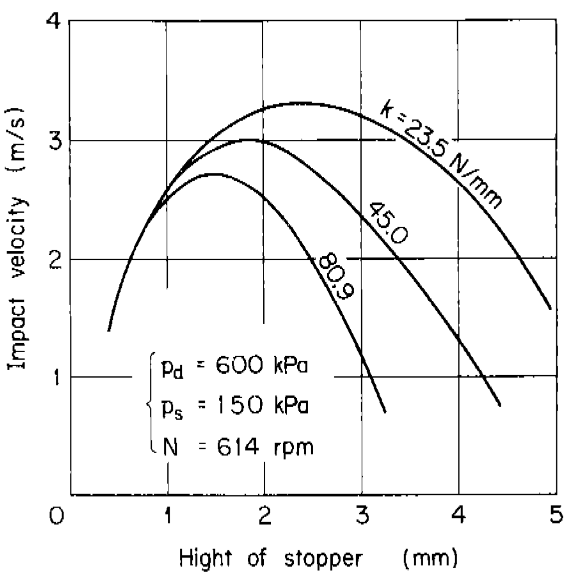


Fig.12 Relationship between impact velocity of discharge ring valve and hight of stopper on parameter of spring constant.

Assuming an impact velocity of 6 m/s for the impact fatigue limit, an impact stress of 244 N/mm^2 is obtained by Eq. (18). This value is very low in comparison with the tensile and flexural fatigue limits and the impact fatigue limits obtained by Svenzon (4) and Dusil(5). This result suggests that further investigations are needed to clarify whether tearing off of chips from edges is induced or not under collinear impact conditions around an impact stress of 250 N/mm^2. However, it seems to be very hard to produce impact fatigue failure under such a small impact stress. Since an impact stress of 250 N/mm^2 is obtained by the collinear contact between the valve and the valve seat or valve stop, the other predominant impact conditions must be taken into consideration to clarify the mechanism of the valve impact failure.

Predominant Impact Stress Affecting Impact Fatigue Failure of Valve

Authors(10)∿(14) have both theoretically and experimentally examined the valve dynamics over a wide range of compressor operating conditions. These results show that the valve does not strike the valve seat or valve stop perfectly squarely, but with flexural mode shape. This mentions that the valve impact does not occur under the collinear impact conditions, but occurs under the oblique impact conditions. The fundamental importance of oblique impact for the creation of impact fatigue fracture has already been mentioned by Svenzon(4). Through this study, it is noticed that its importance was confirmed, and that the treatment as problems of locally impacted beam or ring is important to clarify the creation of impact fatigue of the compressor valve.

CONCLUSIONS

In order to get fundamental knowledge of the valve impact stress, a preliminary impact stress analysis was conducted, then comparing with the test results obtained by the longitudinal impact of two bars. On the bases of these results, the impact duration and maximum impact stress of valves, and the transient response of strain gauges were discussed. Moreover, predominant impact stress affecting impact fatigue failure of valves was also discussed. The results obtained from this study are as follows.

(1) The theoretical solutions of elementary equations for wave propagation are obtained with velocity-impact loadings at the impact end of two finite elastic bars. The theoretical resulsts are in good agreement with the experimental results.

(2) Strain gauges have good transient response for measuring the stress wave propagation in bars with comparatively long length. However, this technique is not successful to measure the impact stress propagation of the compressor valve of which the thickness is thin, because an impact duration is about (0.59 ∿ 7.8) x 10^{-7} s against that the transient response of strain gauge is about 4 x 10^{-6} s.

(3) Under the collinear impact conditions, impact stress is about 250 N/mm^2 when the valve hits the valve seat or valve stop with an impact velocity of 6 m/s. This value is very low compared with the tensile and flexural fatigue limits and impact fatigue limits of valve materials.

(4) Further investigations are considered to be needed to clarify whether tearing off of chips from valve materials is induced or not under collinear

impact conditions around an impact stress of 250 N/mm^2.

(5) The oblique impact of valves against the valve seat or valve stop is important for clarifying the creation of impact fatigue failure of valve materials. In order to attain this, theoretical and experimental treatments as problems of locally impacted beam or ring is important in further investigations.

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