1986

To Reduce Impact Noise of the Valve by Using Silent Valve Plate

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TO REDUCE IMPACT NOISE OF THE VALVE BY USING SILENT VALVE PLATE

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ABSTRACT

Through the sound spectrum analysis, this paper indicates that the noise peak frequency comes from mechanical shock, investigating the frequencies of noise sources, the noise peak frequency is found — the impact pairs of the valve plate and the spring. The cause to raise the noise is on the phenomenon — detachment between the valve plate and the spring. For this reason the valve plate produces high frequency vibration by the external exitation.

The efficent way to reduce the impact noise is to adopt the high damping material. The test using four kinds of material on the compressor L2-10/8-I shows they have decayed the SPL (A) well.

SYMBOLS

- A valve plate area
- C sound wave velocity before jet
- C_o sound wave velocity after jet
- C_R reflecting coefficient of valve plate
- D cylinder diameter, spring diameter
- d steel wire diameter of spring
- E modulus of elasticity

E_{acc} sound energy of acceleration

F impact force

F_r impact force at radial

F_s impact force at axial

f frequency

G shear elasticity

g acceleration of qravity

h valve lift

h_o beforehand compressional deformation of the valve spring

I valve plate cross section moment of inertia

i vibration ordinal numbers, spring ring number

K valve spring constant

k C_p/C_v

L_f sound power spectral level

L_w sound power level

l free height of spring

M Mach number of valve

m mass of valve plate

p pressure in the valve

Δp pressure loss of the valve

r valve plate radius

S piston stroke

T period

t time

V velocity

V_o first impact velocity

V_{vg} impact velocity of the valve plate-guard (or seat)

V_{vs} impact velocity of the valve plate-spring

V_{ps} at the time of impact, the vibration velocity of the valve plate

V_{ss} at the time of impact, the vibration velocity of the spring

W valve plate gravetation

X ampletude of vibration

β pushing force coefficient of the valve

μ unified sound energy flux density of acceleration

ρ density, radiation efficiency of noise

ω angular velocity of compressor, angular frequency of forced vibration

ω_0 angular frequency of free vibration

η decaying factor

INTRODUCTION

The main noise source comes from the valves, it consists of airflow noise, air-flow pulse noise and impact noise. The former two are easy to control, the third is difficult to control. This artical investigates the very problem. Usually people hold that the impact noise mainly comes from the impact of the valve at the valve guard or valve seat and the SPL of the former is the greatest, the peak frequency of which is $f = in/60$ (for single acting $i=1$, double $i=2$), so treating the impact noise as low frequency. However, this paper holds that the impact noise mainly comes from the impact between valve plate and spring, their detachment is of the cause to impact, its peak frequency is high frequency, so we should treat it as high frequency.

THERE IS A PEAK FREQUENCY AT THE HIGH FREQUENCY DOMAIN OF THE VALVE NOISE

Fig. 1 shows the sound spectrum of the first stage valve noise of the compressor 2V-0.6/7 (n=1450 r.p.m). The Fig. 1a shows the spectrum without muffler, the Fig. 1b shows the spectrum with a muffler. The bots have the peak valve at the high frequency domain, especially the one with a muffler is evident (Fig. 1c). The same phenomenon occurs on other compressors. Fig. 2 shows the spectrums of some ordinary compressors with mufflers. The peak frequency at the high frequency domain stimulates the hearing organ being stronger then low frequency. From the noise rating number curve (Fig. 3) we can see that the necessary decaying amount at the high frequency domain is much more greater than that at the low frequency domain. So we should put more attention to the peak of high frequency.

THE PEAK FREQUENCY NOISE COMES FROM THE IMPACT

The peak frequency of air-flow pulse is low frequency f=in/60 (single acting i=1, double i=2), the peak frequency of throttling jet is low or middle frequency[1] and its noise power is a function of Strouhal Sh (Sh=fD/V), the peak frequency at the Sh=0.2[2] (Fig. 4). So the high frequency noise must be from impact noise.

INVESTIGATING THE HIGH FREQUENCY NOISE SOURCE

High freqency impact noise must be produced by high frequency vibration impact pair. There are valve plate-spring system, spring, valve plate to cause vibration in the valve structure.

1. Valve plate-spring system (Fig. 5a):

The pushing force of the air-flow $\beta \Delta pA$, is the external excitation of the valve plate-spring system. Considering no damp, the differential equation

$$\frac{W}{g}\ddot{X} + KX = \beta \Delta pA$$

Where, $\Delta p = \frac{k\pi^2}{8} p(\sin\omega t + \frac{\lambda}{2}\sin 2\omega t)^2 M^2$

Considering the piston movement as simple harmonic motion and combining

$$\sin^2\omega t = \frac{1}{2}(1 - \cos 2\omega t)$$

so

$$\frac{W}{g}\ddot{X} + KX = \frac{\beta k\pi^2 ApM^2}{16}(1 - \cos 2\omega t) \qquad (1)$$

So

$$X = \frac{\beta k\pi^2 ApM^2}{16K}\left[\frac{1}{1-4(\frac{\omega}{\omega_o})^2} - 1\right]\cos\omega_o t + \frac{\beta k\pi^2 ApM^2}{16K}\left[1 - \frac{\cos 2\omega t}{1-4(\frac{\omega}{\omega_o})^2}\right]$$

(2)

The first item represents the free vibration of the systim, the second depends on external excitation. The forced vibration cycle is

$$T = \frac{2\pi}{2\omega} = \frac{\pi}{\omega} \qquad (3)$$

The ω of the piston compressor is smaller, so the vibration frequency is low frequency.

2. Spring, Valve plate

The impact force is the external excitation to cause them free vibration. The detachment between the spring and the valve plate causes the separately free vibration of the spring and valve plate. After impact, the valve plate reflects, at the same time, the impact given to the spring by the valve are transmitting within the spring by shock wave, its cycle is

$$T = \frac{2\pi i D}{V} \tag{4}$$

Where, $V = \frac{d}{D}\sqrt{\frac{G}{2\rho}}$

Considering no damp, the vibration curve of the spring is shown at Fig. 5b, at least the detachment occurs within O-T . The undamped natural frequency of the spring is from (4)

$$f_s = \frac{1}{T} = \frac{d}{2\pi i D^2}\sqrt{\frac{G}{2\rho}} \tag{5}$$

From (5), the natural frequency ot the spring is low frequency. The free vibration frequency of the valve plate consists of radial vibration frequency and torsional vibration frequency. In fact the movement of the valve plate is a tilted motion which would cause a tilted impact, the force of which may be resolved into a radial component Fr and a axial one Fs (Fig. 6). At the act of Fr the valve plate vibrates freely along radial, At the act of Fs the valve plate vibrates in torsion. So there are two vibration frequency [3]

$$f_{pr} = \frac{1}{2\pi}\sqrt{\frac{Eg(1+i^2)}{\rho r^2}} \tag{6}$$

$$f_{ps} = \frac{1}{2\pi}\sqrt{\frac{EgIi^2(1-i^2)}{\rho A r^4(1+i^2)}} \tag{7}$$

Where, i = 1, 2, 3, ...

The free frequency of the valve plate is high frequency. So we consider the valve plate to be the high frequency noise source.

The impact pairs include

1. Valve-guard (or seat):

Their impact velocity

$$V_{vg} = C_R^{i-1} V_o \tag{8}$$

The momentum

$$mV_{vg} = mC_R^{i-1} V_o \tag{9}$$

2. Valve-spring

Their impact velocity

$$V_{vs} = V_{vg} + V_{ps} + V_{ss}$$

Where, V_{ps} — At the time of impact, the vibration velocity of the valve plate.

V_{ss} — At the time of impact, the vibration velocity of the spring.

Considering no damp, the vibration equation of the spring

$$X = (h + h_o)\cos\omega t + \frac{V_{vg}}{\omega}\sin\omega t$$

So, $$\dot{X} = -\omega(h + h_o)\sin\omega t + V_{vg}\cos\omega t$$

At $\omega t = 2(n-1)\pi$, $(n=1, 2, 3, \ldots)$, the impact occurs and at this moment the vibration velocity of the spring

$$V_{ss} = -\omega(h + h_o)\sin(2n-1)\pi + V_{vg}\cos 2(n-1)\pi = V_{vg}$$

Substituting them into the above

$$V_{vs} = 3V_{vg} = 3C_R^{i-1} V_o$$

i.e. momentum

$$mV_{vs} = 3mC_R^{i-1} V_o \tag{10}$$

From (9), (10) we can see that the momentum of the valve-spring mV_{vs} is three times of the momentum of the valve-guars mV_{vg}. The actual tested curve has demonstrated this fact. If the impact momentum of the valve to the guard were greatest, the greates amplitude should have been at the position of first impact. In fact it is not the situation, it has a certain lag (Fig. 7). This explains the greatest impact force does not occurs at the impact of the valve plate to the guard, but at the impact of the valve plate to the spring.

REDUCING THE IMPACT NOISE HIGH DAMPED VALVE PLATE

Through the above analysis, it can be seen that the better way to reduce the noise is to adopt high damped valve plate replaing the metal valve plate. The reason follows

1. The impact noise power reduced:

For the high damped valve plate, its C_R and m are much smaller the metal one. So its momentum mV_{vs} is much smaller too.

2. Sound energy flux dinsity of radiated noise of valve plate vibration redmed:

The sound energy flux density of the radiated noise of thevibration body

$$I = 83\pi f^2 X^2 \rho \cdot 10^{-7} \qquad (W/cm^2)$$

E of high damped meterials are smaller than E of the metal valve plate, from (6) (7) we can see f is much smaller than f of metal. However, the decaying factor η is much greater than that of metal, so its vibretion amplitude X is much smaller than that of metal, consequently its sound energy flux density of the vibration noise of the high damped valve plate is much smaller than that of metal.

3. The energy conversion from momentun to sound reduced:

Fig. 8[4] shows the conversion from impact energy to sound energy. $\mu = E_{acc} / \frac{1}{2} m V_o$ in Fig. 8 represents the unified sound energy flux density. Where, m - the mass of the gas equal to the valve plate volume, V_o - the impact velocity of the valve plate, C - the sound speed, t - the clapsed time reducing V_o to zero. The curve shows that for a given V_o , incrasing t makes reducing guickly and the high damped valve plate can increase t, consequently the energy conversion from impact to sound is reduced. This curve directs us, if δ from 1 to 10, is 0.1 to 0.0001, the noise level is reduced by 30 dB. it is too difficult to reach it for the general way.

4. The transmition of the high frequency noise reduced:

Because of $f_v \gg f_s$, besed on the bibratory theory, the spring relative to the valve plate is "stile rigid body", so the a spring transmited the high frequency without decaying, and the high damped valve plate reduced the transmition, decaying the solid sound radiation.

The valve plates made of four selected high damp material have been used in the compressor L2-10/8-I and tested, the whole noise of the machine is reduced by 2 - 3 dB (A). The noise spectrums of the first stage high damped valve plate and metal valve plate are shown in Fig. 9.

CONSULTING DOCUMENTS

[1]Qian Xinghua, Wang Xiaojing, "An Approach to Reckoning of Peak Noise Frequency of Air Compressor Inlet", "FLUID ENGINEERING", 1986.1. P33-P39

[2]Ma Dayou, "ACOUSTICS DIRECTORY", Science Publesh Society, 1983. P207

[3]S. Timoshenko, "VIBRATION PROBLEMS IN ENGINEERING", Third Edition, D. Van Nostrand Company Inc, 1955

[4]E. J. Richards, "INTER - NOISE 1982 " P43-65

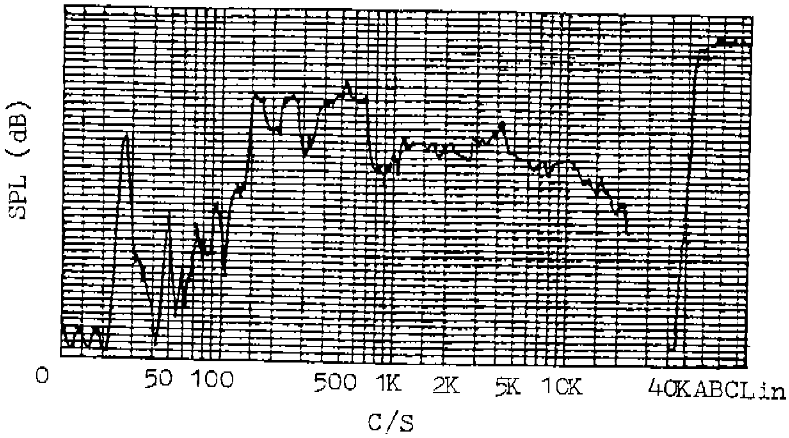


Fig. 1a: Measured Sound Spectrum of the First Stage Valve of the Compressor 2V-0.6/7 Without Muffler.

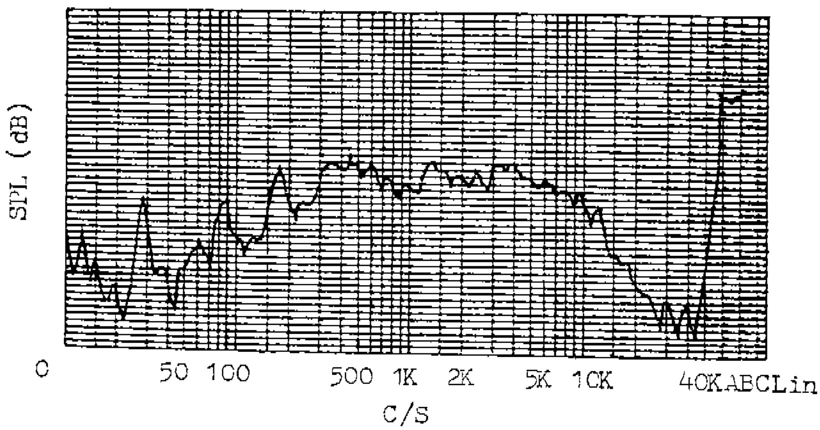


Fig. 1b: Measured Soud Spectrum of the First Stage Valve of the Compressor 2V-0.6/7 with Muffler.

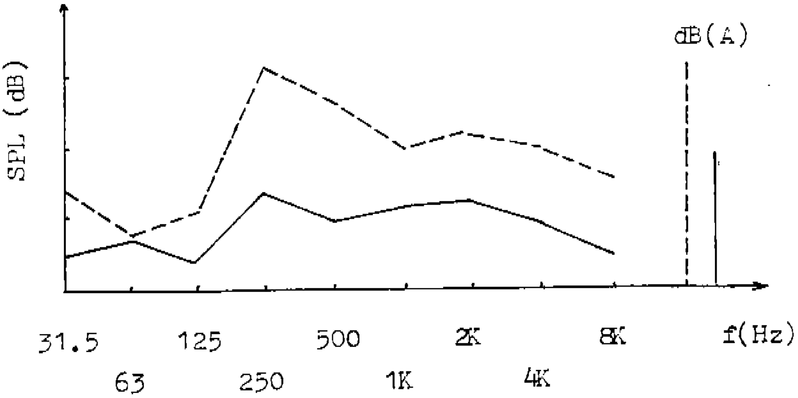


Fig. 1c: Comparison of the Octave Sound Spectrum of the First Stage Valve of the Compressor 2V-0.6/7 Measured Without Muffler (-----), with Muffler (———).

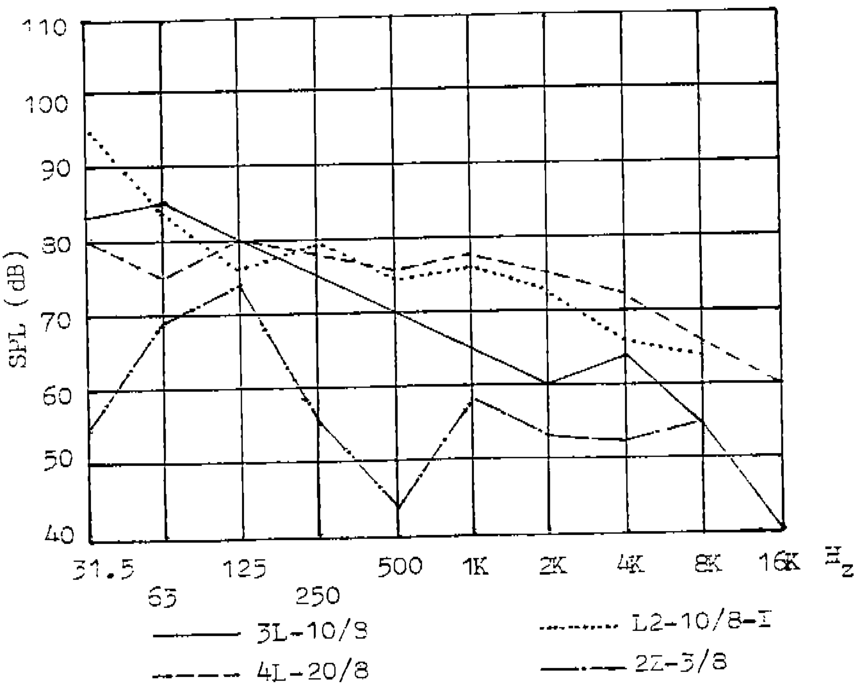
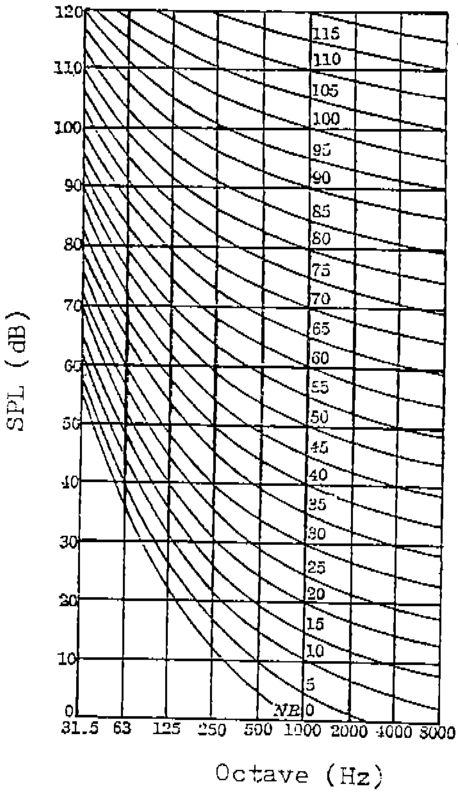


Fig. 2: Measured Octave Sound Spectrums of the First Stage Valve of Some Ordinary Compressors with Mufflers.

Fig. 3: Curve of the Noise Rating Number.



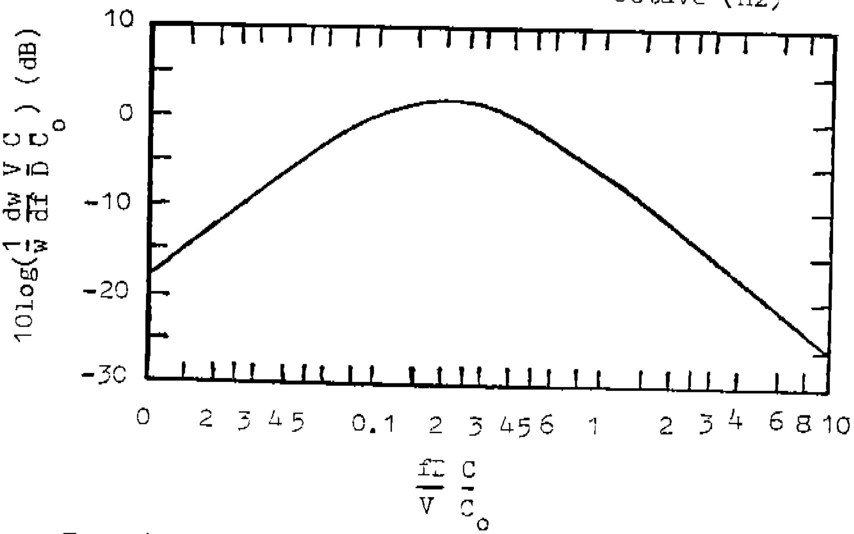


Fig. 4: Unified Sound Spectrum of Jet Noise.

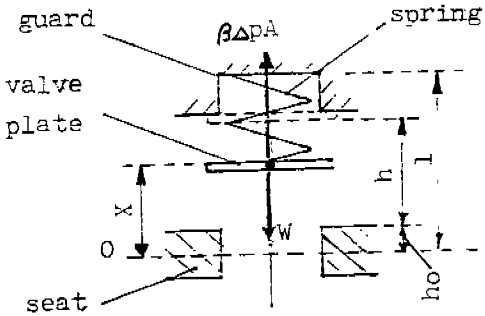


Fig. 5a: Forced Vibration of the Valve Plate—Spring System.

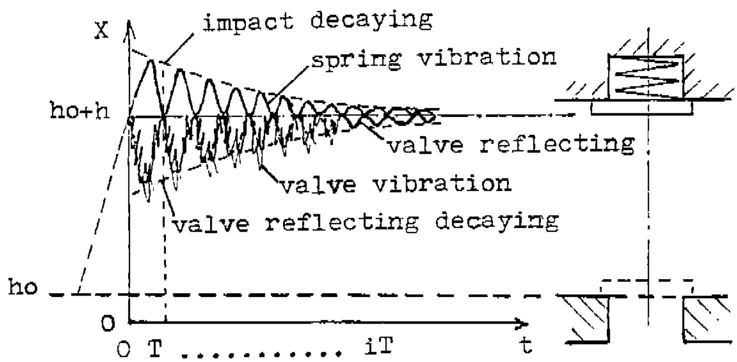


Fig. 5b: Impact of the Valve Plate with Spring, Free Vibration of the Valve Plate and Spring.

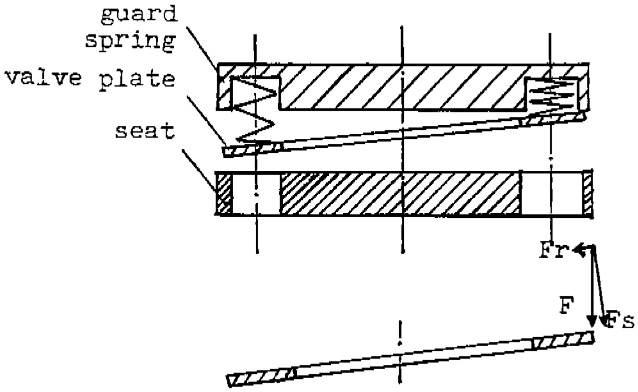
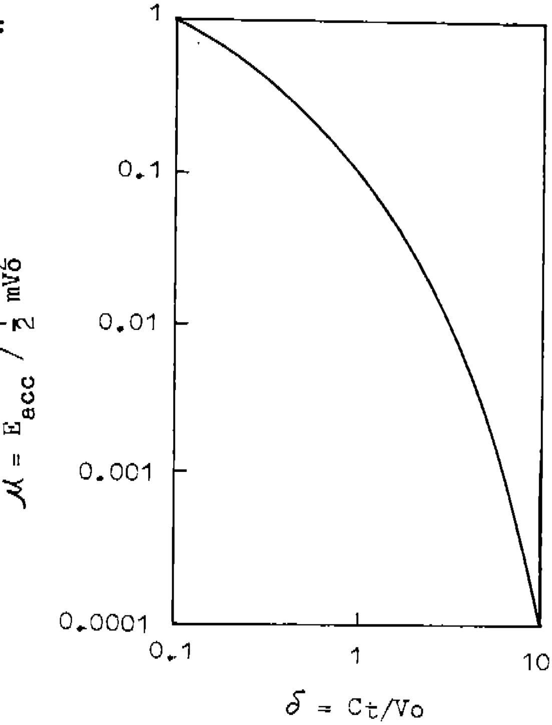


Fig. 6: Resolve of Tilt Impact Foree of the Valve Plate.

Fig. 8:



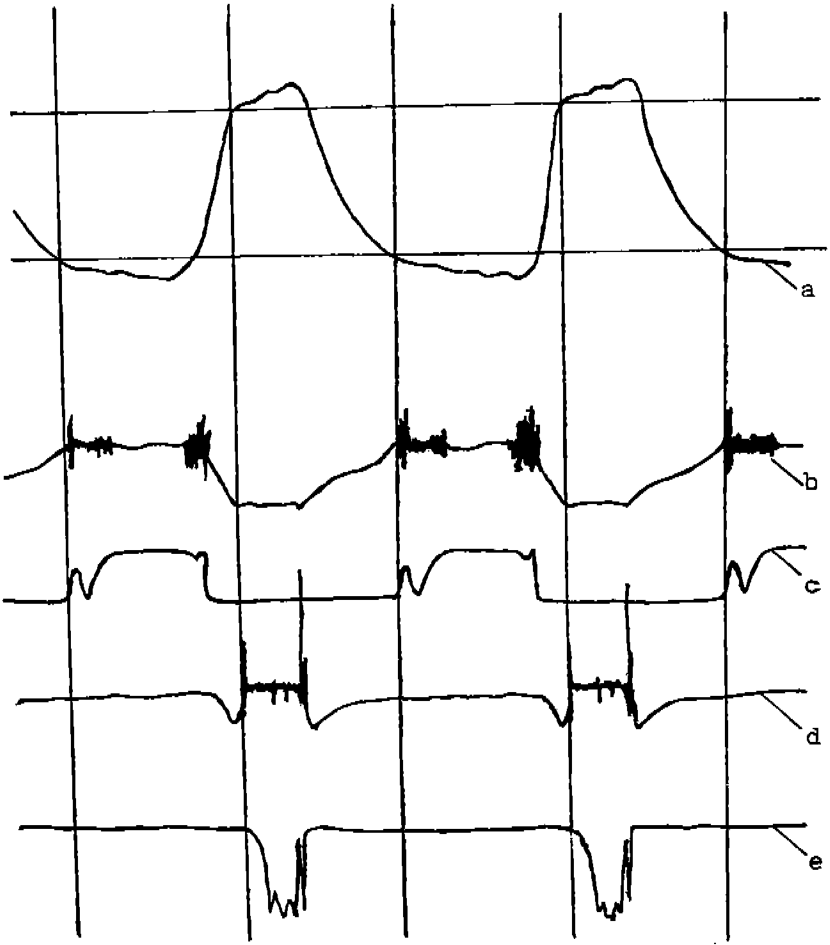


Fig. 7: a. Indicator Diagram.
b. Stress Diagram of Suction Valve Plate.
c. Motion Diagram of the Suction Valve Plate.
d. Stress Diagram of Discharge Valve Plate.
f. Motion Diagram of the Discharge Valve Plate.

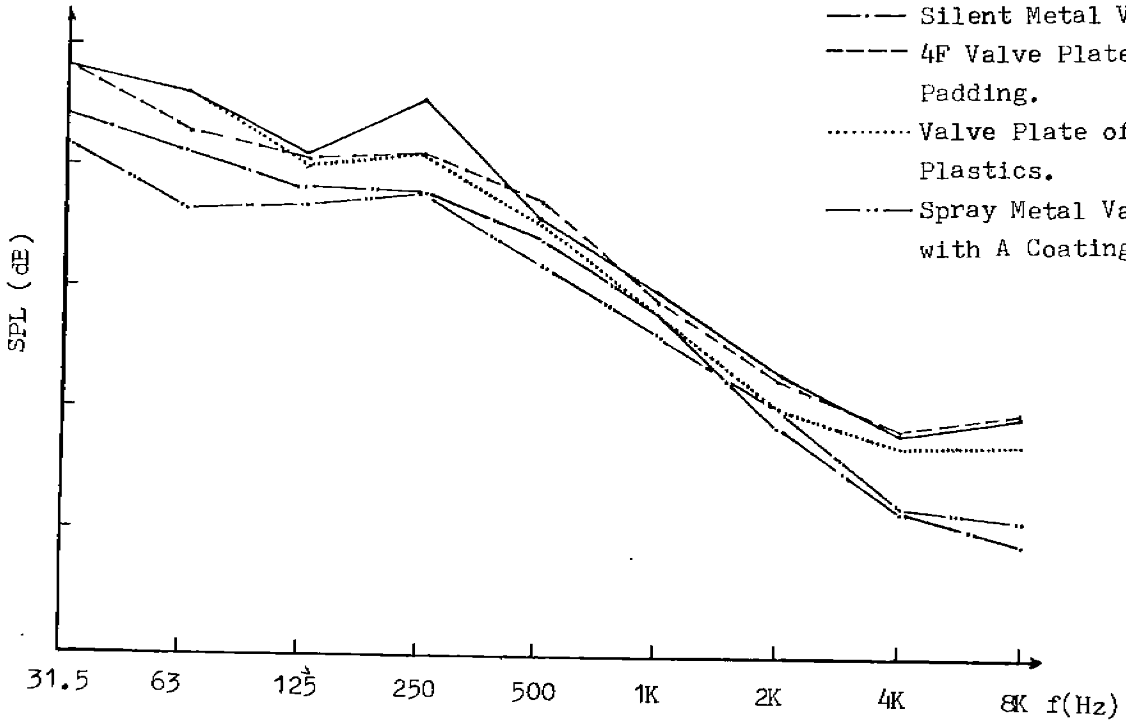


Fig. 9: ----- Metal Valve Plate.
—·— Silent Metal Valve Plate.
— — — 4F Valve Plate with Some Padding.
·········· Valve Plate of A Phenolic Plastics.
—··— Spray Metal Valve Plate with A Coating of 4F.