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SELF-EXCITED VIBRATIONS OF RECIPROCATING COMPRESSOR PLATE VALVES

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ABSTRACT

The self-excitation mechanism causing the flutter of reciprocating compressor plate valves is experimentally investigated under steady state conditions. The relation between the plate oscillations and the resulting pressure pulsations is established by means of simultaneous recording of the plate movements and the upstream and downstream pressure pulsations.

It is shown that movements of the valve plate produce fluid forces which are destabilizing because they are in phase with the velocity of the plate. Thus, acoustic resonances of the piping system are strongly excited due to their coupling with the vibration of the valve plate. The stiffness of the valve springs has been found to have very little influence on the vibration frequency which is controlled by the system acoustical resonances.

Whilst the system oscillation is sustained, the valve plate stays parallel to the seat, but substantial plate tilting eliminates this vibration. Thus, the valve vibrations can be eliminated by using unevenly distributed springs which force the valve plate to tilt while opening.

1. INTRODUCTION

Self-excited vibrations of spring loaded compressor valves are sustained via a destabilizing fluid force which increases as the valve lift is increased. Several investigations have characterized experimentally (1, 2) and theoretically (3, 4) this destabilizing fluid force as a force (or a pressure) coefficient which is a function of the valve lift. However, the nature and the cause of this destabilizing force is not self-evident for all types of valves.

In this paper, we examine a multi-ring plate valve for which the self-excitation mechanism is least understood. In this type of valve, the flow experiences two 90° bends, and the exit jets are very closely spaced. These features make the dynamic behaviour of the valve plate very much influenced by the flow dynamics not only upstream but also downstream of the valve plate. This is in contrast to other valve types for which the valve behaviour is controlled by the flow dynamics only upstream of the valve plate (3, 4).

The multi-ring plate valve, hereafter referred to merely as the valve, is tested under "steady-state conditions" i.e. at constant upstream and downstream pressures. Measurements of the plate vibrations at four locations, the pressure pulsations upstream and downstream of the valve plate, and the phasing of events are conducted for several values of the spring stiffness, the maximum valve lift, and the length of the upstream pipe. These measurements facilitated an assessment of the valve dynamic behaviour under symmetric and asymmetric flow conditions. The cause and nature of the fluid forces which destabilize the valve are also addressed.

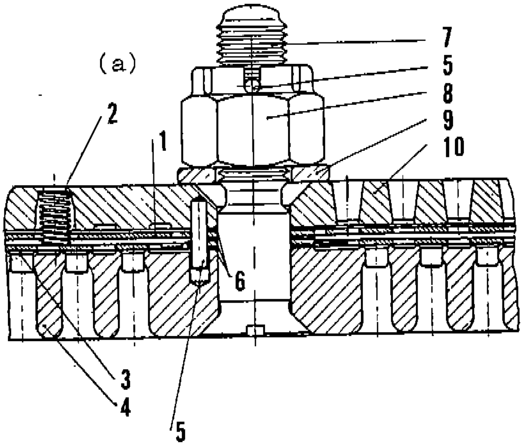
2. EXPERIMENTAL SET-UP

The complete valve assembly, a part of which is shown in Fig. 1.a, was installed at the end of a pipe connected to a pressurized air supply and exhausted into the atmosphere as shown in Fig. 1.c. The air mass flow rate was determined by means of an orifice plate.

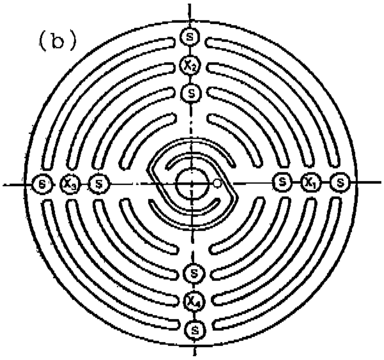
Four proximity transducers type Bentley 1/4" mounted through the valve seat or impact plated and calibrated in situ gave the mean valve lift and the frequency and amplitude of the plate vibrations at the four locations shown in Fig. 1.b. A pressure transducer type Endevco 8510B mounted through the side of the valve seat measured the pressure fluctuations immediately upstream of the valve plate; while another same type transducer in the impact plate gave the pressure variations behind the valve plate. The six signals were recorded simultaneously on a UV recorder to facilitate phase analysis. Frequency analysis of all signals was caried out on-line by means of a real time analyser type Nicolet 444A.

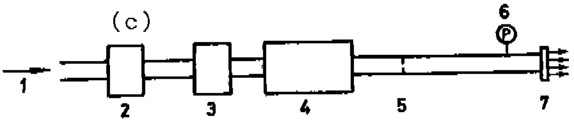
All the tests were conducted without the damper plate. However, some tests were carried out at the end

1: damper plate
2: spring
3: valve plate
4: valve seat
5: pins
6: valve lift spacers
7: bolt
8: locking nut
9: washer
10: impact plate



s: location of springs
x: location of proximity
transducers





1: pressurized air supply 2: filter 3: pressure regulator 4: silencer 5: orifice plate 6: pressure gauge 7: test valve

Figure 1: (a) valve assembly; (b) valve plate showing locations of displacement transducers and springs; (c) experimental set-up.

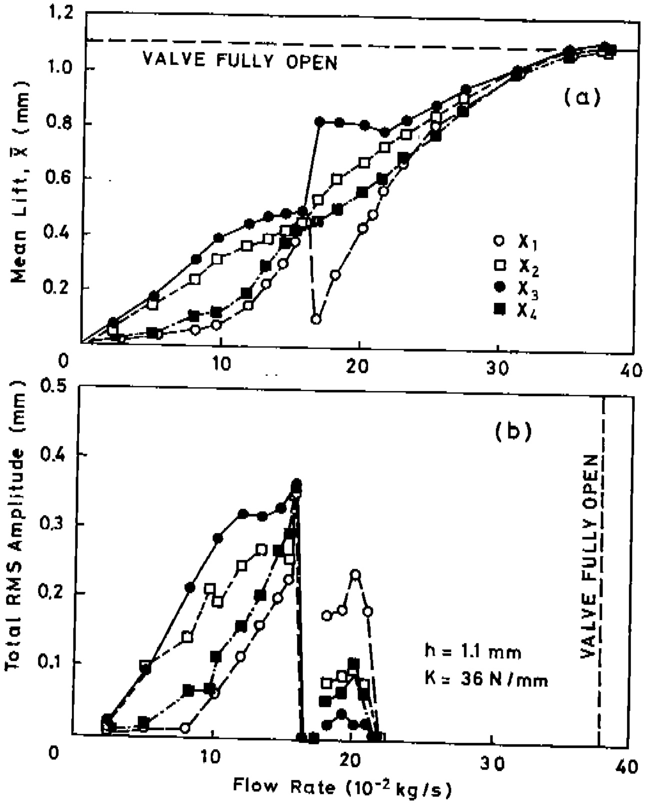


Figure 2: Mean valve lift and total RMS amplitude vs. air flow rate; without damper plate; h = 1.1 mm; K = 36 N/mm.

with the damper plate installed to investigate its effect on the valve vibrations.

3. EXPERIMENTAL RESULTS

3.1. Range of Vibrations

Figs. 2 to 5 show the graphs of the mean valve lift and the total RMS amplitude of vibrations at the four locations shown in Fig. 1.b plotted against the air mass flow rate. In the figures x is the valve opening, h is the maximum valve lift, and K is the retaining spring stiffness. The flow rate was increased by increasing the upstream pressure. In all the tests it was observed that the plate started to vibrate when the mean lift at at least one of the four locations was

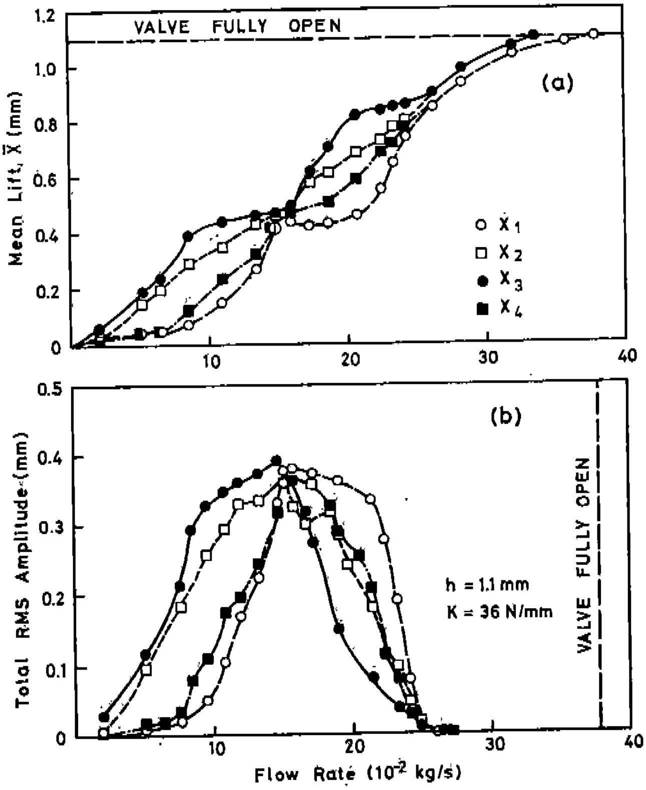


Figure 3: Same as in Fig. 2 but with another set of identical springs.

more than 0.05 mm. The plate vibrations continued throughout the valve opening to about 80% of the total valve lift when they stopped except when considerable tilting of the plate occurred.

3.2. Amplitude of Vibrations

The amplitude of vibrations increases steadily with increasing flow rate (hence with increasing valve lift) up to a maximum value and then decreases to zero with further increasing flow rate as shown in Figs. 2.b to 5.b. The maximum amplitude of vibrations is quite large with the peak-to-peak value being more than 80% of the total valve lift. These strong vibrations can cause severe damage (side wear and breakage) of the springs. Also the valve plate can be damaged at the edge from impacting on the seat and impact plates, and fatigue cracking of the plate can occur near its centre.

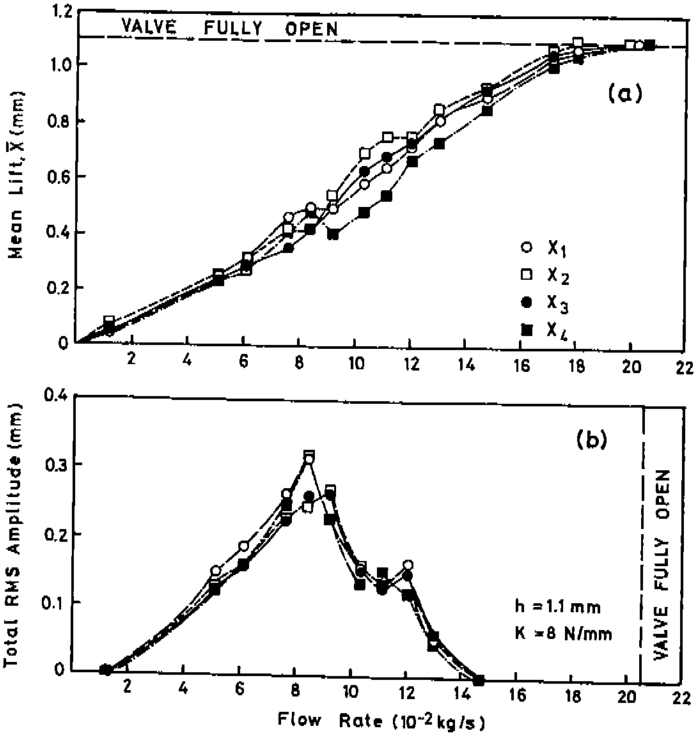


Figure 4: Mean valve lift and total RMS amplitude vs. air flow rate; without damper plate; h = 1.1 mm, K = 8 N/mm.

3.3. Frequency of Vibrations

The dominant frequency of vibrations is observed to change gradually with increasing mass flow rate, i.e. with increasing valve lift. The frequency of all the vibrations shown in Figs. 2.b to 5.b is from about 163 to about 190 Hz. In general, the vibrations started at 190 Hz when both the valve lift and the vibration amplitude were small. Thereafter, the frequency decreased gradually to about 163 Hz when the valve was about half-open and the amplitude was at its maximum. Further increases in the flow rate caused the frequency to increase gradually back to 190 Hz when the vibrations subsided at large valve lifts.

Interestingly, the frequency of vibrations was not altered at all by changing the stiffness of the retaining valve springs from 36 N/mm to 8 N/mm; in both cases the frequency range was from 165 to 187 Hz. However, the amplitude of vibrations was slightly smaller

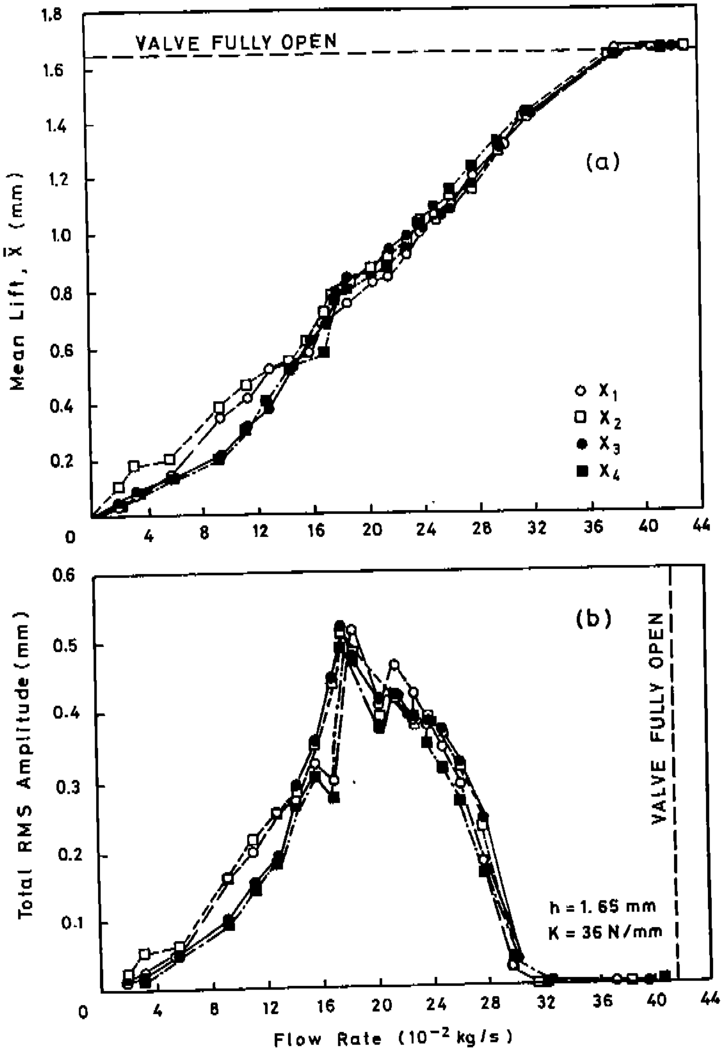


Figure 5: Mean valve lift and total RMS amplitude vs. air flow rate; without damper plate; h = 1.65 mm; K = 36 N/mm.

for the softer springs (Fig. 4.a). The fact that the frequency was independent of the spring stiffness underlined the possibility that the valve vibrations are coupled to some acoustical resonance of the system. This possibility is discussed in the next section.

It is noteworthy that some low frequency vibrations (f = 20 and 58 Hz) were observed occassionally when the main vibrations were not present. Since the intensity and occurrence of these vibrations were much less than those of the main vibrations further tests on these low frequency vibrations were not pursued.

3.4. Effect of the Upstream Pipe Length

Since the vibration frequency was not influenced by spring stiffness, it was decided to design an experiment to check whether or not the valve could be coupled to an acoustical resonance of the system. The resulting test arrangement, Fig. 6.a, made the vibrations more intense because the upstream pipe in this case is much more reactive. Quite clearly the vibration frequency decreases as the length, L, of the upstream pipe is increased, Fig. 6.b. Moreover, not only a single frequency is excited, but a whole series of higher harmonics and subharmonics are. In Fig. 6.c the ratio L/λ vs. L is given, where λ is the wavelength. It is

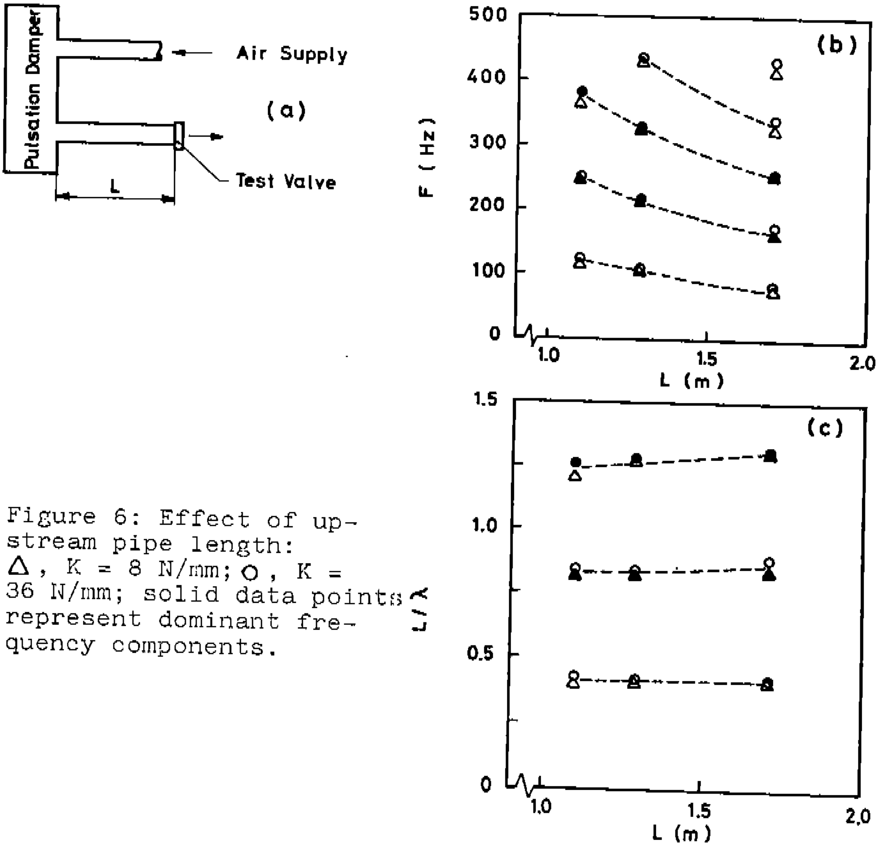


Figure 6: Effect of upstream pipe length: Δ, K = 8 N/mm; ○, K = 36 N/mm; solid data points represent dominant frequency components.

seen that the ratio L/λ of the fundamental resonance mode is slightly smaller than 0.5 which should be expected because the valve plate increases the mass of the system.

The effect of the spring stiffness has also been tested on this system. As shown in Figs. 6.b and 6.c, the vibration frequencies are solely controlled by the length of the upstream pipe; or by acoustic resonances of the upstream pipe. The effect of changing the spring stiffness is confined to selecting which resonance mode would dominate; the higher the stiffness the higher the order of the dominant mode.

3.5. Effect of Plate Tilting on Vibrations

Some tilting of the valve plate always occurs as indicated in Figs. 2.a to 5.a. The maximum amplitude of vibration occurs when the plate is moving parallel to the seat, i.e. no tilting; this is clearly demonstrated by the results shown in Figs. 2 and 3. When considerable tilting occasionally occurs while the plate is vibrating at large amplitude, the vibrations immediately stop; this is shown by the results in Fig. 2. The test conditions of the results shown in

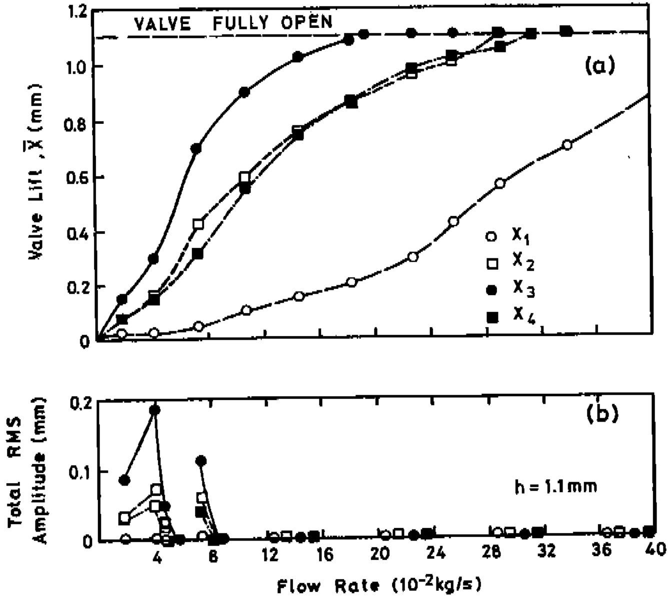


Figure 7: Effect of forced tilting: mean lift and total RMS amplitude vs. air flow rate; K = 9 N/mm at x_1; K = 2 N/mm at x_2, x_3 and x_4.

Figs. 2 and 3 are identical, except that another set of identical springs were used. In the case shown in Fig. 3, plate tilting does not occur and thus the valve vibrations continue with an amplitude gradually decreasing to zero. This different behaviour with practically identical test conditions indicates that plate tilting under symmetrical flow and spring forces is very sensitive; it may or may not occur, and to the authors' opinion it is not controllable. The same behaviour was also observed with the softer springs.

The effect of plate tilting was further investigated by mixing soft and stiff springs so as to force the valve plate to tilt. The results for two values of maximum valve lifts, h = 1.1 and 1.65 mm, are given in Figs. 7 and 8. The tendency of the plate to vibrate is seen to be greatly reduced. Effectively the plate does not vibrate when large tilting occurs.

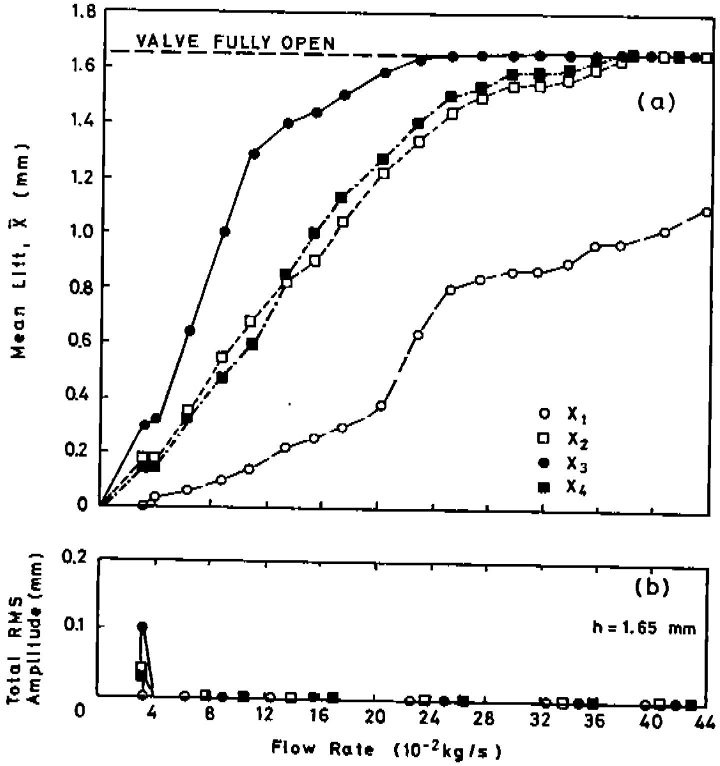


Figure 8: Effect of forced tilting: mean lift and total RMS amplitude vs. air flow rate; K = 9 N/mm at x_1, K = 2 N/mm at x_2, x_3 and x_4.

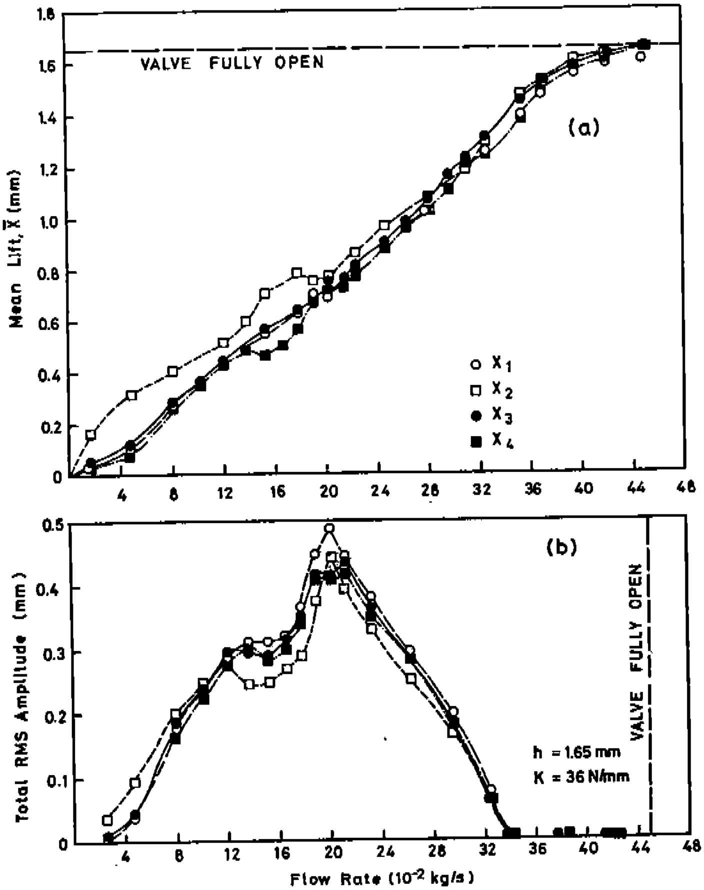


Figure 9: Mean valve lift and total RMS amplitude vs. air flow rate; with damper plate; h = 1.65 mm; K = 36 N/mm.

3.6. Effect of the Damper Plate

Some tests with several combinations of spring stiffness and valve lift were repeated but with the damper plate installed. The results of one of these tests with the conditions similar to those of Fig. 5, are given in Fig. 9. Comparison of these two figures indicates that the damper plate has virtually no effect on the amplitude nor frequency of vibration.

4. DISCUSSIONS

4.1. Excitation Mechanism

Fig. 10 shows typical time traces of the upstream pressure, the downstream pressure, the pressure difference, and the plate vibrations at four locations. It is seen in Fig. 10.c that the plate movements at the four locations are in phase, indicating not only that the mean position of the plate is parallel to the valve seat as has been shown earlier in Figs. 2.a and 5.a, but also that the plate movement is purely translational. This means that the flow field is, rather surprisingly, axisymmetric and the resulting fluid forces are in phase everywhere.

Considering the phase between the upstream pressure and the plate displacement, i.e. P_u and x_3 in Fig. 10.b, it is seen that during the opening part of the cycle, P_u decreases from its maximum to minimum value, whereas it increases from its minimum to maximum value during the closing part of the cycle. This behaviour is not surprising even when it is not considered as a result of the pressure waves in the upstream pipe. When the valve opening increases, the pressure just upstream of the valve should decrease because the flow losses in the upstream pipe increase and also because the flow must accelerate. On the other hand, when the valve opening decreases, the flow losses decrease and the flow must deccelerate; this results in an increase in the upstream pressure. The important observation to underline here is that a movement of the valve plate produces upstream pressure variations which force the plate back to its initial position. Therefore, the upstream pressure has a stabilizing effect.

The behaviour of the downstream pressure is generally similar to that of the upstream pressure, but since the downstream pressure acts on the back side of the plate, it has a destabilizing effect. As shown in Fig. 10.b, the downstream pressure, P_d, is at its maximum when the valve opening is at its minimum, and vice versa. This can be explained by referring to Fig. 10.a, which shows a portion of the valve terminated at both sides by a streamline. Since the gap width, t, is constant, the flow velocity in this gap is linearly related to the valve lift. As the valve lift increases, the flow velocity in the gap will also increase and thus the back pressure, P_d, will decrease.

The last time trace to consider is the pressure difference, ΔP, across the plate, see Fig. 10.b. It is seen that ΔP during the opening part of the cycle is

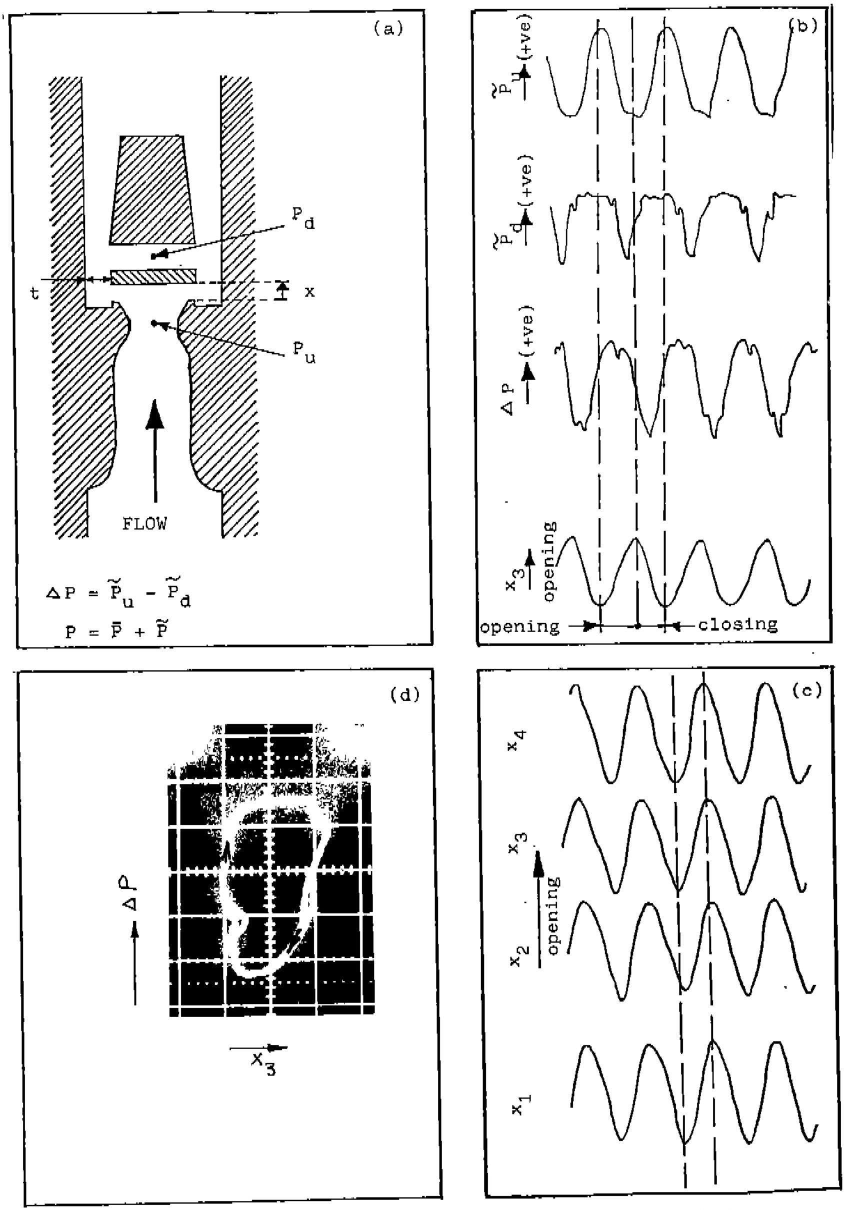


Figure 10: (a) Flow path in a valve. (b) and (c) Pressure and displacement time traces. (d) Pressure drop vs. displacement representing net energy transfer from the fluid to the elastic structure.

considerably higher than that during the closing part. This implies that the work done by the fluid on the elastic structure (i.e. the springs) during the opening part of the cycle is larger than that done by the structure on the fluid during the closing part of the cycle. The result is that there is a net energy transfer from the fluid to the structure during each cycle. This net energy which perpetuates the oscillations is represented by the bounded area shown in Fig. 10.d which presents the pressure difference as a function of the valve lift.

The most interesting feature of the time trace of ΔP shown in Fig. 10.b is that the variations of the pressure difference are virtually in phase with the vibration velocity of the plate; i.e. maximum positive pressure difference occurs at maximum positive plate velocity, and maximum negative pressure difference occurs at maximum negative plate velocity. This behaviour is referred to as a negative damping whose destabilizing effect has been well investigated.

4.2 Effect of Tilting on the Excitation Mechanism

The test results show that the system oscillations are due to a coupling effect between the valve vibrations and the system acoustical resonance whereby the plate movements provide a feedback link which accentuates the acoustical resonance. This feedback mechanism seems to be at its strongest when the valve plate moves parallel to the seat. As soon as the valve plate experiences considerable tilt, the vibrations immediately subside. This suggests that the feedback mechanism can be self-sustained only when the valve plate remains parallel to the seat. In other words, tilting of the valve plate seems to destroy this feedback mechanism. Since tilting of the valve plate cannot have that much influence on the system acoustical resonance, to explain the effect of the plate tilting one has to clarify the valve dynamic behaviour under symmetrical and asymmetrical flow conditions.

As mentioned earlier, when the plate stays parallel to the seat, the fluctuations of the back pressure are all in phase (axisymmetric case). Integration of these fluctuations would then result in a strong component of negative damping which destabilizes the system. On the other hand, if plate tilting occurs the flow dynamics in the valve channels are no longer symmetric. Upstream of the plate, the flow will be biased to one side and downstream of the plate the flow will be attached to one side of the downstream diffusers. Thus the pressure pulsations at the back of the plate would be no longer in phase and would cancel each other out.

Under these conditions, the feedback mechanism which perpetuates the system resonance cannot be self-sustained. Although this seems to be the only explanation to the effect of the plate tilting on the excitation mechanism, further tests including flow visualization on a large scale model are currently underway so as to provide more insight into the flow dynamics inside the valve channels. The results of the flow visualization will be reported upon in a future paper.

4.3 Cause of Plate Tilting

It is not clear why large tilting of the valve plate occurs under symmetrical flow and retaining spring forces. However, when it does occur it is about a particular axis which may be a preferred axis of tilting due to the geometry of the plate near the centre where it is clamped. For the valve used in this investigation x_2-x_4 is the preferred axis of tilting. The shape of the plate near its centre is such that tilting about x_2-x_4 axis is easier than about the x_1-x_3 axis.

Another reason which may cause tilting under symmetrical conditions is that the spring forces from the nominally equal size springs may not be perfectly balanced when the valve is in operation. Since the springs are free to shift somewhat in their recesses in the impact plate, the direction of the individual spring forces can change slightly as the plate vibrates thereby inducing tilting. Tilting of spring loaded plate valves under symmetrical flow and spring forces has also been reported upon in (5) and analysed in (3).

4.4 Means of Reducing Plate Vibrations

This investigation has shown that large forced tilted motion of the valve plate caused by mixing of soft and stiff springs results in much less oscillations of the plate. This could be a practical way of reducing such vibrations in actual compressors. However, the effect of tilted motion on impact stresses would have to be investigated first. The advantage of impact under forced tilted motion is that the exact positions of impact are known; and necessary precautions to alleviate the effects of this impact can effictively be taken as has been suggested by some researchers (6). Here, one should recall the fact that the plate will tilt anyway under symmetrical spring forces. In this case the plate tilting will occur randomly and necessary counter-measures to reduce the impact stresses cannot be taken.

Other ways of reducing the plate vibrations, e.g. by using an optimum combination of spring stiffness, initial spring force, and maximum valve lift, or by increasing the system damping, are currently being investigated and will be reported upon in future.

5. CONCLUSIONS

The self-excited oscillations of a multi-ring plate valve have been investigated. The valve oscillations are coupled to an acoustical resonance mode of the upstream pipe. Changes in the spring stiffness of the valve do not have any influence on the vibration frequency which is determined by the system acoustical resonance.

It is shown that the pressure pulsations upstream of the valve are stabilizing and those downstream of the valve are destabilizing. The net effect of these pressure pulsations is a fluid force which is in phase with the velocity of the valve plate. This is equivalent to a negative damping which destabilizes the system. The main contributing source to this negative damping is the pressure fluctuation downstream of the valve.

The excitation mechanism can be self-substained only if the valve plate stays parallel to the seat at all times. Therefore, the vibration amplitude reaches its maximum when the valve plate is parallel to the seat, whereas the vibration subsides immediately when the plate tilting becomes considerable. It is also shown that tilting of the valve plate under symmetrical donditions may or may not occur, i.e. it cannot be controlled.

The valve vibration can be eliminated (or reduced) by forcing the valve plate to tilt while being opened. This can be achieved by mixing soft and hard springs. Since this approach predetermines the impact locations, counter-measures to reduce the impact stresses can be adopted.

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