Flow Forces and the Tilting of Spring Loaded Valve Plates - Part III

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FLOW FORCES AND THE TILTING OF SPRING LOADED VALVE PLATES
Part II

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ABSTRACT

Using the basic relations from Part I the investigation of stability of seat parallel motion is extended from line loads to springs acting as point loads. Correction factors are given which take into account geometric arrangement of springs.

Stability of motion for real conditions during opening and closure is discussed. A ball-groove model is given for a better understanding of tilted motion. Measurements of tilted motion published by MacLaren are discussed and found to support author’s reasoning. Recommendations are given for preventing instability and for estimating stability from steady state flow tests.

SPRINGS ACTING AS POINT LOADS

Up to now we have assumed spring load to be distributed along the center line of the channel length. Now we consider springs acting as discrete forces at given points. As a simple example we take the strip valve plate of fig.6, now with only 2 springs acting symmetrically. A tilting disturbance shifts the acting line of the resulting spring force \( F_{spr} \) outside the center line.

The same is true for the impulsive force, fig.12.

Valve motion obviously remains stable, if \( F_{spr} \) is shifted more than \( F_i \), i.e.

\[ M_{spr} > M_i \] (14)

Here \( M_{spr} \) stands for the moment resulting from spring forces, and \( M_i \) for the moment resulting from impulsive forces. For small inclination angles \( \alpha \) both these moments are proportional to \( \alpha \). Therefore the amount of \( \alpha \) does not affect stability.

For the moments we can write

\[ M_{spr} = \sum_{i=1}^{n} \Delta F_{spr,i} \cdot z_i = \sum_{i=1}^{n} \Delta y_i \cdot z_i = \frac{1}{n} c_{spr} \cdot \sum_{i=1}^{n} z_i^2 \] (15)

In this equation \( c_{spr} \) is the stiffness of an individual spring and \( c_{spr} \) is the overall stiffness; \( n \) denotes the number of spring forces (index \( i \) for the \( i \)-th spring). For the moment resulting from the impulsive forces we get

\[ M_i = \int (\partial F_i / \partial y)_{yo} \cdot \alpha z \cdot ds \cdot z = \alpha (\partial F_i / \partial y)_{yo} \int z^2 \cdot ds \] (16)

\( (\partial F_i / \partial y)_{yo} \) denotes the gradient of the

\[ \text{FIGURE 12} \quad \text{Springs acting as point loads} \]
impulsive force (per unit length of channel) with respect to valve lift \( y \), taken for the value \( y_0 \), for which stability is investigated. For the simple configuration fig. 2, we find (see appendix, Part I)

\[
\frac{\partial F_i}{\partial y} y_0 = 0.78 \Delta p \tag{17}
\]

(independent from lift \( y_0 \))

The quantities \( \sum z^2 \) and \( \int z^2 \cdot ds \) are closely related to the moment of inertia in dynamics and the same transformation rules can be applied.

If many springs are regularly distributed in distances \( \Delta s \) eq (14) reduces to eq (12). This can be easily shown (\( l = n \cdot \Delta s \)):

\[
\frac{c_{spr \cdot i} \cdot \sum z^2_i}{n} \approx c_1 \sum z^2_i \cdot \Delta s = c_1 \sum z^2_i \cdot \Delta s \tag{18}
\]

Having this in mind, we can use all the equations and procedures given previously in connection with distributed spring force, when \( c_1 \) (overall spring stiffness for unit length of channel) is multiplied by a correction factor \( f \) taking into account geometric configuration of spring loads:

\[
f = \frac{1}{R_m} \int \frac{\sum z^2_i \cdot \Delta s}{z^2 \cdot ds} \tag{19}
\]

The integral has to be performed along the center line of all channels of the valve.

From analogy to the theory of moments of inertia it arises that all configurations with more than 2 axis of symmetry have equal stability irrespective of angular position of tilting axis. Table 5 gives some values for simple cases. From this table it also arises that there is practically no difference between discrete spring load for valves with 1 ring. For a typical 3-ring plate valve with springs acting in the medium ring, we find \( f = 0.7 \) and hence stability is reduced when compared with distributed spring load.

**TABLE 5**

<table>
<thead>
<tr>
<th>Correction factors ( f ) according to equation (19)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Ring with 3 Springs</strong></td>
</tr>
<tr>
<td><img src="image" alt="3-ring plate valve" /></td>
</tr>
<tr>
<td>( f = 1 )</td>
</tr>
<tr>
<td>( f = 12 \frac{a^2}{l^2} )</td>
</tr>
<tr>
<td>( \frac{R_m}{R_a} \cdot \text{medium Radius of outer ring} )</td>
</tr>
<tr>
<td>( R_i = 0.5 , R_a )</td>
</tr>
</tbody>
</table>

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STABILITY OF MOTION AND TILTING OF VALVE PLATE

Tilting can result from unsymmetry in flow and/or spring conditions, irrespective of stability. But tilting may occur also with completely symmetrical flow and/or spring conditions, when the valve motion is unstable. If this occurs, and to what degree, depends on random disturbances on the degree of instability and on the time interval, in which plate travels through an unstable zone etc.

Typical conditions are as described below. When opening, plate accelerates in stable zone and passes unstable zone with appreciable velocity. When closing, plate starts with zero velocity at guard, coming immediately into an unstable zone (or passes a short zone of stability only). Acceleration is slow compared with opening. (Closing times are usually several times longer than opening times!). Flow and spring asymmetries are usually more accentuated when the valve is open. Reinisch [6] e.g. found considerable unsymmetries in springs of bending arm type. So we can expect that tilted plate positions are more likely to arise during closing process. Fig. 13 demonstrates this with a ball-groove model.

Let us now confront our theoretical approach with experiments. It is not easy to measure exactly tilted motions of a valve plate. Fortunately MacLaren et al. [7] published an excellent paper in the '78 Conference, which includes valve plate inclination measurements. Those were computed from the output of 3 displacement transducers measuring the lift of a ring plate. From additional valve data, which the author kindly got in a letter from J.P.T. MacLaren, the value \( c_1 \) is calculated to about 1.25 lb/in\(^2\) (0.085 bar). For fully open valve there is \( y/b \approx 1 \). Though the valve has only one ring, the channel arrangement is similar to a multi-ring valve with 2 x 90° deflection flow. However, the characteristic curve of the impulsive force is difficult to estimate and may range between those given in fig. 4 and appendix of Part I. For this reason the following comments include some uncertainty.

Fig. 14 reproduces MacLaren's results as far as it is relevant to our discussion. It must be noted that these measurements were performed to detect tilting due to eccentric arrangement of suction valve with respect to cylinder center line. Point C had the lowest flow intensity.

With respect to stability one could comment these results as follows.

**Speed 550 rev/min**

Opening: From beginning we can see a slightly increasing inclination. Later in the opening process inclination increases rapidly up to 0.4 degrees. The rapid increase in inclination is likely to be connected with instability. Fig. 15 shows this diagrammatically (see also Table 4 in Part I).

![Figure 13: Ball-groove model](image)

![Figure 15: Stability diagram (schematic)](image)
Plate enters instability zone at $\Delta p \approx 0.8$ lb/in². For reasons discussed in connection with fig.13, the inclination does not reach its maximum value.

Closure: Plate starts ($y/b \approx 1$) with some unsymmetrical flow forces at $\Delta p \approx 1$ lb/in² under stable conditions but enters soon instability zone at lower values $y/b$. This leads to maximum inclination of 1.8 degrees. For small $y/b$ motion becomes again stable and inclination remains low.

Speed 420 rev/min

$\Delta p$ in general is lower. Flow force is not sufficient to press valve plate to guard (condition of motion can be expressed by symbol "Pa", Table 3 of Part I,--plate oscillates). When the plate touches instability zone at low relative lifts $y/b$, motion becomes unstable and this leads to maximum inclination angle.

To the author's opinion unsymmetrical flow conditions in this case are responsible only for the initiation of the tilting motion, which is highly amplified due to instability.

CONCLUSIONS

- Tilted motion of valve plate can occur even when flow is strictly symmetrical. The phenomenon is accessible to theoretical treatment. Knowledge of flow force and spring characteristics are necessary for that.
- The development of large pressure differences across valve (irrespective if caused by flow resistance or by gas pulsations in piping) and low spring stiffness are governing the conditions for instability of valve motion and consequent tilt.
- If valve dynamics is calculated by computer, simple additional calculations ($\phi/\Delta p=f(y/b)$) can predict stability of valve motion. These calculations are recommended (provided that realistic data concerning flow forces is used!).
- With advanced computer programs tilted motion could be calculated in detail, starting with a nominal tilting disturbance of say 0.1 degrees when entering instability zone.
- Springs for valves should be selected not only with respect to correct closing time of valve, but also with respect to stability of valve motion. High spring stiffness is favorable. Coil springs with high precompression and only little increase in force with valve lift are very disadvantageous (small stiffness!).
- Symmetrical conditions for flow and spring load should be sought carefully. Irregularities are greatly amplified when unstable conditions of motion apply.
Springs of a given stiffness located at greater distance from plate center give considerable better stability (proportional to square of distance).

The instability phenomenon discussed occurs also when making steady state flow experiments; here parallel equilibrium position within the limits of valve lift becomes unstable. As stability for a given configuration depends on the dimensionless quantity

\[ \frac{f \cdot c_i}{\Delta p} \]

one has to apply pressure differences \( \Delta p \) as actually in the compressor. If compressed air supply to the steady state test rig does not develop the pressure differences wanted, one can reduce spring stiffness \( c_i \) or \( f \) (e.g. by locating the springs nearer to the center!). In many cases, guard does not affect stability and can be removed during steady state stability tests. Stability also is not affected by absolute spring load, hence one can adjust the different valve positions by adding weights. High pressure valves, for which few measurements for plate motion exist, can be investigated in this way at atmospheric conditions.

One could try to get stability also with an impulsive force characteristic, which is constant or decreases with lift \( \partial F_1 / \partial y < 0 \). Some authors in the past used a so-called "drag force" \( F_D = c_D \cdot \Delta p \) with constant \( c_D \), based on rough experiments. But normally there is \( \partial F_1 / \partial y > 0 \) and often investigators noticed instability effects when making steady state tests (even with low values \( \Delta p \)). It may, however, occur with more complicated channel configurations, that \( \partial F_1 / \partial y < 0 \). This would mean inherent stability for all pressure differences \( \Delta p \). But the author feels very strongly that such channel configurations are associated with excessively high pressure losses. The reader interested in the problem of impulsive force is referred to some papers in the preceding Proceedings [8,9,10,11].

We have discussed in this paper instability of motion of stiff valve plates. Similar mechanisms may cause torsional vibrations in reed valves.

REFERENCES