1972

Designing Compressor Valves to Avoid Flutter

R. W. Upfold
Wollongong University College

DESIGNING COMPRESSOR VALVES TO AVOID FLUTTER

R. W. Upfold, Senior Lecturer, Department of Structural Engineering,
Wollongong University College, Wollongong, Australia.

INTRODUCTION

Continuous monitoring of the motion of inlet valves of reciprocating compressors over some 4000 sequential cycles have shown that a mild degree of varying flutter exists, the onset of flutter being somewhat random. From theoretical considerations, it is shown that flutter can generally be checked for. If both operating and valve characteristics are then kept well away from this "danger zone" then considerably better operating efficiencies should, and generally do, result.

From Costagliola's (1) first analytical modelling of compressor valves, many (2,3) have advanced the art of using digital computers to simulate valves, until today, the technique is being refined to account for relatively minor but still important facets such as valve "sticktion" effects, dynamic damping and energy absorbtion on impact. Each of these latter facets present considerable problems when endeavouring to verify the analytical trends on prototype compressors.

During experimental investigations into valve sticktion and dynamic damping, valve motion diagrams were found to be non-reproducible on later occasions. Three inductance type gauges were employed for monitoring the inlet valve motion of one cylinder of an 8 cylinder industrial compressor. Although the three gauges showed tilting of the valve to a limited degree, with time, the diagrams varied slightly.

To trace the reasons for this time variable motion, a large number of cycles of valve motion were monitored sequentially. The output from each of the three gauges used to monitor the motion were amplified and connected to a relatively high speed data logger. The data logger was in the form of a Nova computer with analogue to digital converters for up to 16 channels. The maximum sampling and storing rate for the logger was of the order of 16,000 per second. This high rate enabled each inlet valve motion diagram to be monitored at 160 points for six continuous cycles (results stored in core) down to 8 points for 4000 continuous cycles (results stored on magnetic tape). Variations between the values stated were also used with the aid of a pulsed starting technique.

After each series of tests the stored output was printed out and also fed through a digital to analogue converter to a Y-time recorder at a much reduced time scale. From a study of such sequential plots it became obvious that from the commencement of operation of a compressor, oil sticktion at the inlet valve seat delays the commencement of valve opening, also reported by MacLaren and Kerr (4). However, it was noted that apart from oil sticktion effects a mild degree of valve flutter existed intermittently.

The valve flutter was not consistent during the test, but commencing each 15 to 25 cycles and then being damped out over the following 12 to 15 cycles. On occasions the variation extended over a period of some 50 cycles. This valve flutter could not be traced to any pressure fluctuations or pulsations in the fluid flow. From an analysis of the results of the three monitoring points, the onset of the mild degree of flutter could not be traced to tilting of the valve or any jamming tendencies.

However, it is conceded that the valve flutter could be due to the inlet valve used (see Fig.1) having a rotary motion not picked up by the three gauges, and thus seating differently. Rotary motion of the inlet valves, to any degree, has not been observed for any series of tests conducted to date. These tests lead the author to believe that the type of inlet valve tested is more prone to flutter than is generally thought. To investigate this, valve flutter was further investigated.

In the tests conducted, little valve flutter existed with the inlet valve lifting where there is generally a very low rise time with the valve under substantial positive pressure forces. With the valve descending back to its closed position, the drop time is generally considerably in excess of the rise time and flutter usually occurs.

It has been found from experiment that the motion of valves of the type shown in Figure 1 is generally as shown in Figure 2, but frequently is of the type shown in Figures 3, 4, 5 and 6. When the motion is as shown in Figure 3, serious damage often occurs as a result of high impact velocities. Generally, the fluctuations are not as severe as shown in Figure 3 where the valve hits both upper and lower stops. Quite frequently the valve leaves the upper seat and keeps fluttering without going far away from the outer stop. Motions of the type as shown by Figures 4 and 5, although not seriously affecting the life of valves, can easily lead to fatigue damage if the valve is worn or damaged from other sources. The motion of the type as shown by Figures 4 and 5 is, of course, highly undesirable from other considerations, such as being far from the optimum shape of the valve lift diagram.

Most results to date follow the general form of Figure 6, where the valve either sticks to the seat or flutters mildly against the seat.

In view of the results obtained, it appears obvious that flutter of the inlet valve contributes greatly to inefficiencies of the inlet valve of reciprocating compressors of the type shown in Figure 1.

Extending the theoretical model developed by Costagliola and further developed by MacLaren, to include dynamic damping, valve sticktion and rebound of the valve, two fundamental equations can be found to describe the pressure across the valve and also its motion, both in terms of each other and the geometric and operating characteristics of the compressor.

In very simplified form, the flow and motion equations for inlet valves are of the form

$$\frac{dX}{d\theta}(\theta,B) = \frac{\beta_1 F(X)B - kZ(\theta)X}{\beta_2 - f(\theta)} \qquad \ldots (1)$$

and

$$\frac{d^2B}{d\theta^2}(\theta,X) = \frac{U(1-X) - B}{C} \qquad \ldots (2)$$

where :

X	pressure ratio;
B	lift ratio;
θ	crank angle;
β	numerical constant;
k	ratio of specific heats;
F	function of k and X;
Z	function of θ, and n;
n	connecting rod/crank radius, and
U and C	are as defined in the text.

In equations (1) and (2), terms incorporating damping, heat transfer and initial compressive force in the spring have been omitted for clarity.

An analytical solution of (1) and (2) is impossible and a numerical solution is used to obtain a Fourier expansion of the simple type,

$$X = a - b \sin d\theta \qquad \ldots (3)$$

where a, b and d may be determined from a least squares fit of experimental results.

Combining equations (2) and (3), we may write

$$\frac{d^2B}{d\theta^2} + \frac{B}{C} = \frac{U}{C}(1 - a + b \sin d\theta), \qquad \ldots (4)$$

which has the general solution

$$B = A \cos\left(\frac{\theta}{\sqrt{C}} + \psi\right) + U(1-a) + \frac{Ub}{1 - d^2C} \sin d\theta \qquad \ldots (5)$$

where A is a constant and provided that $d^2C \neq 1$.

The first term on the right hand side of equation (5) is the complementary function and gives the natural frequency of the system, while the remainder of (5) is the forced oscillation having the same period as X appearing in the differential equations.

The period of the natural frequency is

$$\frac{P_T}{\sqrt{C}} = 2\pi . \qquad \ldots (6)$$

It follows from (4) and (6) that $d^2C \ll 1$ and that d approaches unity.

If the effect of the superimposed natural oscillation is neglected when the valve is fully open (B = 1) then from (5)

$$1 = U(1-a) + Ub \sin d\theta . \qquad \ldots (7)$$

Hence, for the fully open position,

$$\sin d\theta = \frac{1}{b}\left(\frac{1}{U} - (1-a)\right). \qquad \ldots (8)$$

Equation (8) shows that as parameter U increases, crank angle decreases. It follows since

$$\sin d\theta \leq 1$$

that $\frac{1}{b}\left(\frac{1}{U} - (1-a)\right) \leq 1$, or alternatively

$$\frac{1}{bU} \leq \left(\frac{1-a}{b} + 1\right)$$

and $U \geq \frac{1}{(1-a)+b} = \frac{1}{1-(a-b)}$.

Hence $(a - b) = X_c$, a critical pressure ratio which is not equal to 1, and so the result is

$$U \geq 1/(1 - X_c).$$

When $U = 1/(1 - X_c)$ the time for the valve to lift is $\pi/_{2d}$ radians and the valve will be in its extreme position (B = 1). It will return to its seat at $\pi/_d$ radians. It will also be seen that when U = 1, the valve will not lift fully to its extreme position. This is the case for values of $0 < U < 1$.

The actual equations originally derived, incorporating all variables including damping, were solved by first using a Taylor Series expansion and then programming the resulting equations in Fortran IV.

Data on actual valve characteristics were determined from an experimental rig and used in the mathematical model so as to simulate the compressor behaviour. The validity of the mathematical model was investigated and proven approximately correct for particular operating conditions by comparing both experimental and numerical results. The mathematical model was then used to study the motion of the inlet valve and pressure ratio across the valve for a number of combinations of operating characteristics.

Although the approximate analytical solution derived from Equations (1) and (2) does not take into consideration heat transfer, initial compression in the springs or damping, the period of natural frequency derived from equation (6) is in good agreement with the period of natural frequency obtained from the numerical results given by the computer programme.

From equation (6), the period of natural frequency is

$$P_T = 2\pi\sqrt{C},$$

where

- C — $M_v\,\omega^2/K$;
- M_v — mass of valve;
- ω — rotational speed and
- K — spring stiffness.

Shown in Figure 7 are calculated values of P_T for various values of C. Only one area on this Figure has been compared experimentally - this was an oscillation occurring just after the valve lifted and stayed against its outer stop for a short period. An experimental value determined was in the range 0.11 to 0.15 for an analytical value of 0.23 radians when parameter C was 0.0022. As can be seen on Figure 7, the numerical results (shown dotted) compare favourably with the results derived from equation (6). With the value of C equal to 0.05 and 0.5, respectively, the period of natural frequency (greater than 1.4 radians) is too large to be observed in the computer results.

The change in crank angle θ for the duration of the initial valve lift (from $B = 0$ to $B = 1$), may be estimated, using equation (8) rewritten as

$$\theta = \sin^{-1}\left[\frac{1}{b}\left(\frac{1}{U} - (1 - a)\right)\right] \quad . . . (10)$$

However, from the computer calculations for the case where $U = 40$, the inlet valve, commencing at a crank angle of 0.6 radians, reaches the upper seat (where $B = 1$) at 0.6282 radians. Other values are shown plotted on Figure 8.

Using equation (10) and substituting calculated values for $10 < U < 40$ from Figure 8, values of a and b may be found. (See equation (3) for the definition of a and b).

Thus, for the cases where $C = 0.0003$, it is found that

$$a = 3.53$$

$$\text{and } b = 4.35.$$

Using these values of a and b, the calculated values for $B = 1$ compare favourably with those previously obtained. Shown in Figure 9, are values of a and b for other values of parameter C.

The relationships between a, b and C from the calculated values are approximately

$$a = 1.7\,C^{-0.1}$$

$$\text{and } b = C^{-0.2}.$$

Substituting these values in equation (10)

$$\theta = \sin^{-1}\left[C^{0.2}\left(\frac{1}{U} - (1 - 1.7C^{-0.1})\right)\right],$$

and hence $\theta_\ell = \sin^{-1}\left[C^{0.2}\left(\frac{1}{U} - (1 - 1.7C^{-0.1})\right)\right] - \theta_i$

. . . (11)

where

- θ_ℓ crank angle of duration of lift, and
- θ_i crank angle when pressure ratio is unity.

When the lifting period, as defined by equation (11), exceeds half the period of the natural frequency the valve will oscillate when lifting. That is, when

$$\sin^{-1}\left[C^{0.2}\left(\frac{1}{U} - (1 - 1.7C^{-0.1})\right)\right] - \theta_i \geq \pi\sqrt{C},$$

then oscillations occur.

To find a more direct relationship (but still approximate) for the critical combination of U and C where oscillations commence, the relationship shown can be equated and the sine transferred.

That is to say,

$$C^{0.2}\left(\frac{1}{U} - (1 - 1.7C^{-0.1})\right) = \sin\left(\pi\sqrt{C} + \theta_i\right).$$

. . . (12)

From equation (12), a critical value of U is approximately

$$U = \frac{C^{0.2}}{C^{0.2} - 1.7C^{0.1} + \sin(\pi\sqrt{C} + \theta_i)} \quad . . (13)$$

For the particular parameters used in the numerical solution, equation (13) appears to be satisfactory. A more simplified relationship between U and C for the critical conditions (when $\theta_i = 0.6$) is of the form

$$U = \frac{1}{3\sqrt{C}}. \quad . . . (14)$$

This critical combination of U and C, although not producing the shortest lifting period, does yield the fastest lift without oscillations occurring during the cycle. Such a critical combination allows the valve to lift and come into contact with the upper seat with a minimum impact velocity. It permits a combination of values which would enable the impact velocity on striking the upper seat to be rendered almost zero.

By definition, $U = \frac{C_{pd} A_v p_1}{K H_m}$

and $C = \frac{M_v \omega^2}{K}$,

where

C_{pd} coefficient of pressure drag;
A_v area of valve;
p_1 pressure in intake chamber, and
H_m maximum lift of valve.

Equation (14) then gives a combination of design parameters, say A_v, K, H_m, C_{pd} and M_v for operating conditions ω and p_1. An approximate relationship is of the form

$$\omega p_1 = \frac{K^{1.5} H_m}{3 C_{pd} A_v M_v^{0.5}} \qquad \ldots (15)$$

that is, if the product ωp_1 is equal to or less than the right hand term of equation (15) then oscillations or flutter of the inlet valve would occur. Good design of inlet valves for reciprocating compressors would be to ensure that the value of ωp_1 is preferably never less than twice the calculated value of the parameter on the right hand side of equation (15).

In conclusion, it has been found experimentally that intermittent flutter does occur over a relatively large range of operating conditions. Most experimenters have previously not had facilities available for almost continuous recording of valve lift diagrams. This facet has, in the author's case, led to endeavours to force simulated solutions of valve lift upon actual results with frequent mis-match, especially in the "fluttering" areas of the valve lift diagram. It has now been found that variable flutter occurs and indeed other areas of the valve lift diagram may ultimately be found to be variable.

However, apart from this, it has been shown that operating conditions can be designed approximately such that flutter generally does not occur.

REFERENCES

1. Costagliola, M., "The theory of spring loaded inlet valves for reciprocating compressors". Jnl. Appl. Mechs., Dec. 1950, p.415.

2. MacLaren, J.F.T., "Valve behaviour in a small refrigerator compressor using a digital computer". Jnl. of Refrigeration, 11 No. 6 June 1968.

3. Wambsganss, M.W. and Cohen, R., "Dynamics of a reciprocating compressor with automatic reed valves". Proc. X11th Int. Congr. Refrign. Madrid 1967, p.791.

4. MacLaren, J.F.T. and Kerr, S.V., "Automatic reed valves in hermetic compressors". Proc. Comm. III. Int. Inst. Refrign. Prague 1967, p.79.

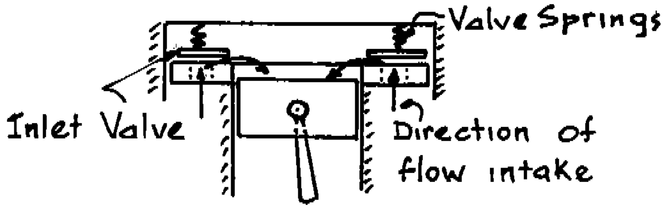


FIGURE 1- Restrictive Inlet Valve

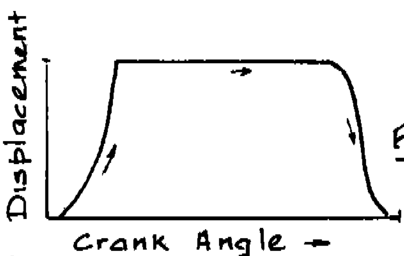


FIGURE 2

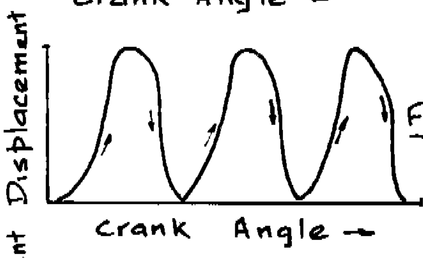


FIGURE 3

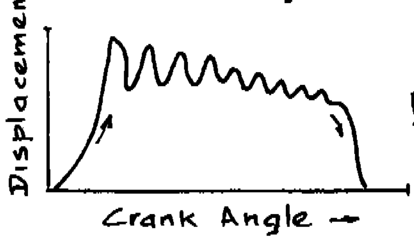


FIGURE 4

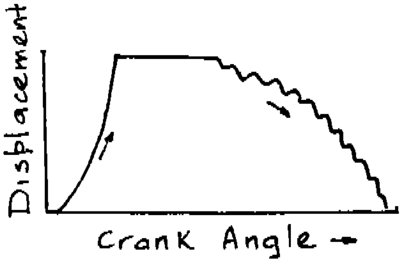


FIGURE 5

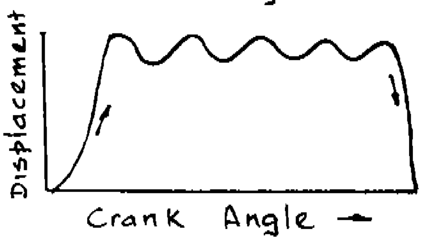


FIGURE 6

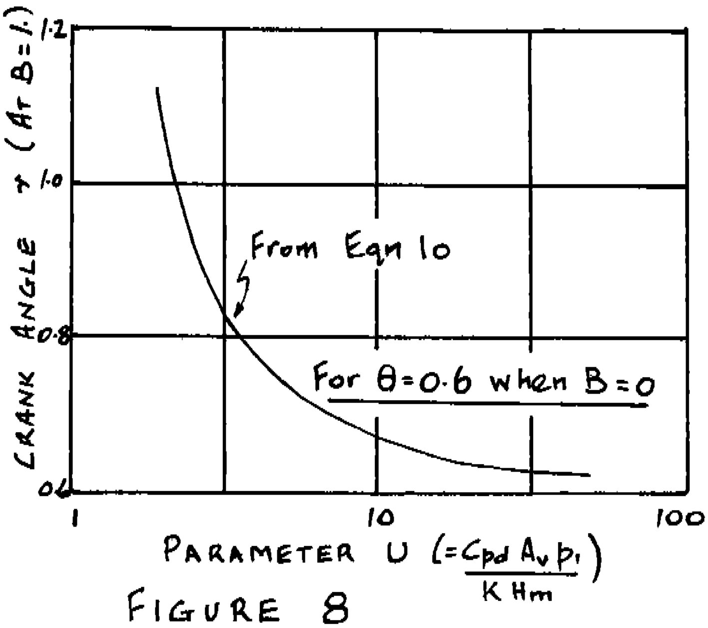


FIGURE 8

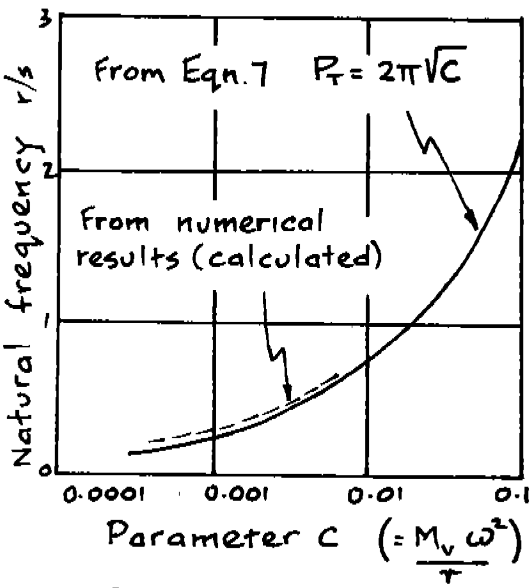


FIGURE 7

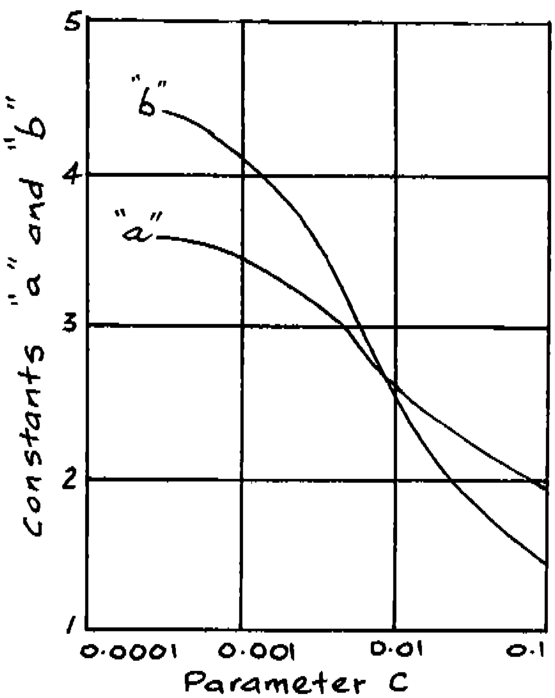


FIGURE 9

DESIGNING COMPRESSOR VALVES TO AVOID FLUTTER

by

Upfold, R.W., Wollongong, University College, Australia

EFFECT OF PIPELINE PULSATIONS ON VALVE FLUTTER

Comments by:

John F. T. MacLaren, University of Strathclyde, Glasgow, U.K.

One of the many aims of mathematical modelling of compressors and their valves is to establish criteria which indicate whether valve flutter takes place. Upfold has used a model in which it was assumed that the discharge line and suction line pressures remain constant, i.e. that no significant pressure pulsations were present.

Brablik (1) investigated the problem of valve flutter using a more complete analytical model which accounted for the pulsations inherent in the system in an attempt to relate the type of valve plate movement with the magnitude of the pressure pulsations, $\frac{\Delta P}{P_o}$. (The effect of the damper plate on the valve plate was neglected in the computations.)

The movement of the valve plate (suction or discharge) was computed by Brablik assuming constant pressure (P_o) in the valve plenum chamber and comparison was made with the movement of the valve plate computed assuming different pressure pulsation amplitudes (1% and 3%), provided by piping of different lengths. This was repeated at different levels of mean pressure, P_o.

At low values of mean pressure, P_o, (region A, Figure overleaf) the spring force (specific loading of the valve plate) was of the same order of magnitude as the forces due to pressure pulsations. This region was found to be one of frequent occurrence of flutter, particularly at low gas flow velocities and low gas densities. The valve therefore "malfunctioned" due to the effect of the pulsations, but neither the number of flutters nor the point of final valve closure was governed by the pulsations in a precise way.

When the mean pressure, P_o, was high (region C) the effect of pressure pulsations on the valve plate movement was significant. At low amplitudes of pulsation a rapid opening of the valve to its maximum permitted lift occurred and the valve remained open against the stop until almost the end of the suction or delivery process. At higher amplitudes, where the amplitude of the pulsations exceeded the pressure drop across the valve, a reversed flow could occur, particularly during valve closure, with the possibility of multiple "slamming" of the valve on the seat (and likely reduction of valve life). The argument was developed further to show that reversed flow was more likely to occur with suction valves than with discharge valves.

In the region B, (intermediate line pressure) both self sustained flutter and a pronounced influence due to pulsation could be encountered, but the valve plate movement depended also on other variables such as the mass of the valve plate.

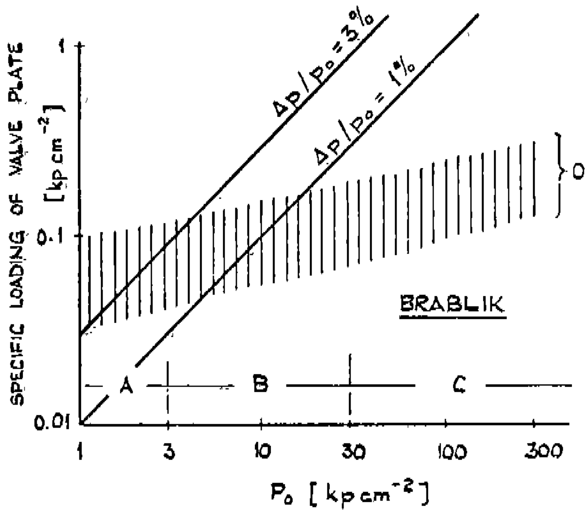
Although pulsations may be considered undesirable the extent of any harm may, in practice, differ considerably in different circumstances even when the amplitude of the pulsations was the same. MacLaren and Tramschek(2) suggested that valve "malfunction" during closure was more sensitive to the phasing of the pulsation than to its amplitude.

Nevertheless, rough guidelines such as presented by Brablik in the figure overleaf, and proved from practical experience to be valid, provide useful information to the valve designer. This approach is reminiscent of the simple method described by Davis (3) to provide approximate criteria for satisfactory valve life.

1. Brablik, J. "Analysis of movement of valve plate of automatic valves under pulsations of gas in the piping of reciprocating compressor". Thesis State Research Institute for Manufacture of Machines, Bechovice, Prague, Czechoslovakia.

2. MacLaren, J.F.T. & Tramschek, A.B. "Prediction of valve behaviour with pulsating flow in reciprocating compressors" A.S.H.R.A.E./ A.S.M.E./Ray W. Herrick Laboratories. Compressor Technology Conference, Purdue University, July 1972.

3. Davis, H. "Effect of reciprocating compressor valve design on performance and reliability" I.Mech.E. Symposium, "Reciprocating and Rotary Compressor Design and Operating Problems" London, Paper No. 3 Oct., 1972



Po = valve plenum chamber mean pressure

Δp = amplitude of pressure pulsation.

O - region of specific loading of valve plate by springs, calculated for 1 cm² of cross-section of channel from equation

$$O = \frac{s\left(h_o + \frac{h_{max}}{2}\right)}{f_k}$$

where s = total rigidity of springs

h_o = assembled compression of springs

h_{max} = compression of springs with valve open

f_k = area of channels