'Pop' safety valves: a compressible flow analysis

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Pressure versus lift characteristics of Pop-type safety valves have been predicted from a model in which pressures on various parts of the valve disc can be deduced from compressible flow analysis of three passages in series. In particular, it has been shown that the characteristic snap-open and snap-shut phenomena are inherent in the solutions to the sets of equations, and are critically dependent on the setting of the adjusting ring, bearing out what is known in practice. The model has shown satisfactory agreement with experimental data from a $\frac{1}{2}$" (12.55 mm) pop valve operating in air.

Keywords: *fluid flow, compressible flow, safety valve, computer model*

Safety valves are essential for vessels containing pressurised fluids; valve types, construction and testing are covered in appropriate Standards[1-3]. A brief history of safety valve development, as well as some limited analysis, has been given by Pearson[4] and an outline of the development of American codes given by Harrison[5].

One type of safety valve, originally developed for use on steam locomotives but now widely used, is the 'Pop' valve, so called because of its action to snap open when the pressure reaches or slightly exceeds the set pressure, and to snap shut when the pressure has reduced. The advantages of this type of valve are that high lift (and hence large discharge and quick pressure control) is obtained rapidly and that the valve can shut tightly after a relatively small drop in pressure.

Many types of 'Pop' valves have been produced, each incorporating a different method of adjusting its operating characteristics. All, however, have a small chamber (called a 'huddling' chamber) around the outside of the valve seat, the discharge area from which can be adjusted (Fig 1). Some valves also have additional control over the back pressure produced within the valve body.

The general shape of the pressure-lift curve, and the effect of the adjusting ring position on over-pressure and blowdown are well known from experience. Analysis has centred either on computing the discharge rate using a steady state model in which the valve is treated as a single orifice (eg Ref 6) or on predicting the dynamic behaviour of high pressure valves, again treating the valve as a single, but variable-area, orifice (eg Ref 7).

The purpose of the work reported here was to develop an analysis which was more appropriate to medium pressure valves (as used on many air receivers, for example), and to take account of the several passages within the valve through which the fluid flows.

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Analytical model

The valve operation is usually rapid, but it is possible to simplify the problem, for the purposes of this work, by making a quasi-steady-state analysis in which flow conditions and the forces on the valve disc are determined for a given lift. The direction of the resultant force then indicates the direction in which the disc would move and by repeating analysis at different lifts, an equilibrium position can be established.

Flow conditions

Because both the valve seat and adjusting ring have seat widths which are finite but small compared with the diameters, the flow through these passages approximates to one-dimensional frictional adiabatic compressible Fanno-typeflow between parallel plates. This can be analysed in a manner similar to compressible flow in circular-sectioned pipes[8] and only the final equations are presented here.

Referring to the passage in Fig 2, the flow into the passage is treated as isentropic, so:

$$\frac{P_1}{P_0}=\left(1+\frac{\gamma-1}{2}M_1^2\right)^{\gamma/(\gamma-1)} \tag{1}$$

For the parallel portion of the passage, the pressure ratio and the effect of friction can be expressed in terms of the Mach numbers as:

$$\frac{P_2}{P_1}=\left(\frac{M_1}{M_2}\right)\left\{\frac{2+(\gamma-1)M_1^2}{2+(\gamma-1)M_2^2}\right\}^{1/2} \tag{2}$$

and:

$$\frac{4fL}{2x}=\frac{1}{\gamma}\left\{\frac{1}{M_1^2}-\frac{1}{M_2^2}\right\}+\frac{\gamma+1}{2\gamma}\times\ln\left\{\left\{\frac{M_1}{M_2}\right\}^2\frac{2+(\gamma-1)M_2^2}{2+(\gamma-1)M_1^2}\right\} \tag{3}$$

For given geometry, stagnation and downstream pressures, and assumed friction factor, these equations define the flow.

The mass flow rate is:

$$\dot{m} = A_1\rho_1 u_1 = A_1\left\{\gamma p_0\rho_0 M_1^2 \Big/ \left(1+\frac{\gamma-1}{2}M_1^2\right)^{(\gamma+1)/(\gamma-1)}\right\}^{1/2} \tag{4}$$

In general, as the downstream pressure is reduced, both M_1 and M_2 increase (with $M_2 > M_1$) and a point will be reached when $M_2 = 1$. The passage is then said to be 'choked' and the downstream pressure has no further effect on the flow.

For isentropic flow in a conventional convergent nozzle, the mass flow rate is given by:

$$\dot{m} = A\left\{\frac{2-\gamma}{\gamma-1}\, p_0\rho_0\left[\left(\frac{p_1}{p_0}\right)^{2/\gamma} - \left(\frac{p_1}{p_0}\right)^{(\gamma+1)/\gamma}\right]\right\}^{1/2} \tag{5}$$

This becomes a maximum when $p_1/p_0 = \{2/(\gamma+1)\}^{\gamma/(\gamma-1)}$, and again further reduction in downstream pressure has no effect on the flow.

Fig 3 shows the pop valve approximated as two passages and one nozzle in series, the first passage being at the valve seat, the second at the adjusting ring seat, and the nozzle being at the discharge from the valve body.

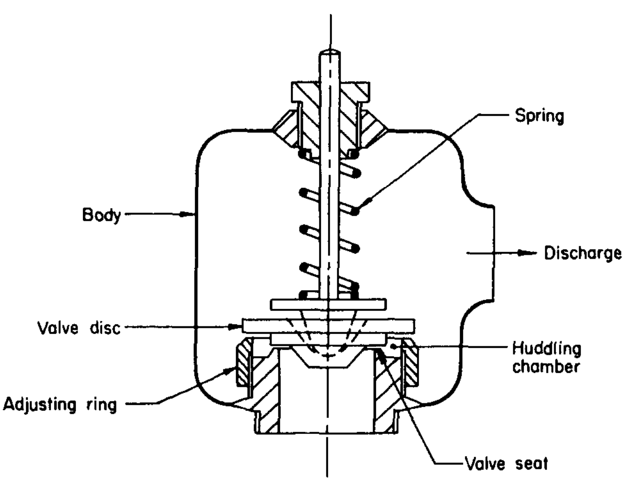


Fig 1 Main features of a 'Pop' safety valve

Each will be governed by the appropriate expressions given above; in addition, by the assumption of adiabatic flow, the enthalpy at the intermediate stagnation conditions will equal that of the initial condition. This, combined with the necessary equality of mass discharge rates, enables the intermediate stagnation pressures to be determined.

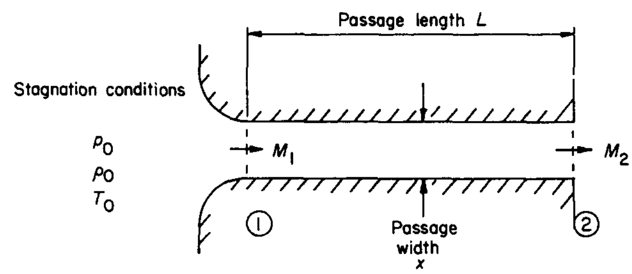


Fig 2 Parameters relating to a convergent nozzle and a parallel passage

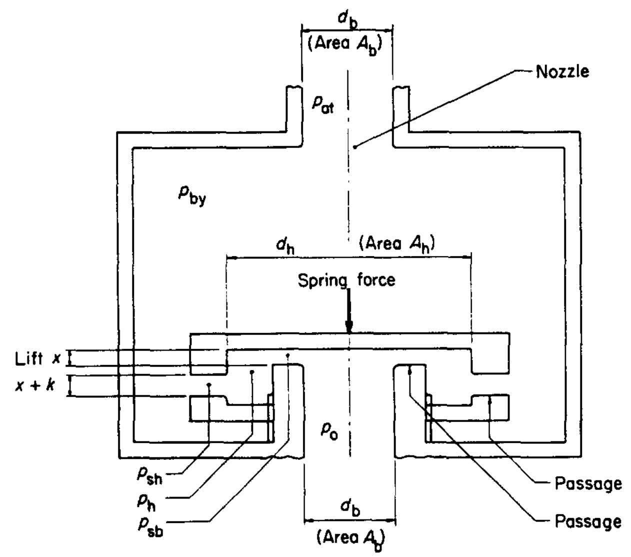


Fig. 3 Idealized Pop valve for compressible flow analysis

Notation

A	General flow area	M_1	Mach number at section 1
A_1	Flow area at section 1 (or for passage 1)	M_2	Mach number at section 2
A_2	Flow area for passage 2	p_0	Stagnation or applied pressure
A_b	Valve bore area projected on valve disc	p_1	Static pressure at section 1
A_{sb}	Valve seat area projected on valve disc	p_2	Static pressure at section 2
A_h	Area to outer edge of huddling chamber projected on valve disc	p_{sb}	Mean pressure within passage formed by valve seat
A_{sh}	Adjusting ring seat area projected on valve disc	p_h	Stagnation pressure in huddling chamber
d_b	Valve bore	p_{sh}	Mean pressure within passage formed by adjusting ring seat
d_h	Huddling chamber outer diameter	p_{by}	Pressure in valve body
f	Friction factor	p_{set}	Set pressure
F_i	Initial force in valve spring	p_{at}	Atmospheric pressure
F_{up}	Net upward force on valve disc	S	Spring stiffness
F_{down}	Net downward force on valve disc	x	Valve lift
k	Initial gap at adjusting ring	γ	Ratio of specific heats of working fluid
L	Passage length (= width of seat)	ρ_0	Stagnation density of fluid
$\dot{m}$	Mass rate of flow	ρ_1	Density of fluid at condition 1

Force balance

Referring to Fig 3, the upward and downward forces on the valve disc are:

$$F_{up} = p_0 A_b + p_{sb} A_{sb} + p_h A_h + p_{sh} A_{sh} \quad (6)$$

$$F_{down} = p_{by}(A_b + A_{sb} + A_h + A_{sh}) + F_i + S \,.\, x \quad (7)$$

Here p_{sb} and p_{sh} are the mean pressures in the two passages formed by the valve seat and by the adjusting ring. It is possible to calculate the pressure profile in these passages and so calculate a true mean. However, this considerably increases computational time, and it was deemed sufficiently accurate here to take the mean as the arithmetic average of the inlet and outlet pressures in the parallel portion, ie referring to the passage in Fig 2, $\bar{p} = (p_1 + p_2)/2$.

F_i is the initial force in the spring, controlling the pressure at which the valve just begins to lift. That this is not just equal to $(p_{set} - p_{at})A_b$ can be seen from Fig 2 by considering the trend in pressures as the lift x tends to zero. Then p_1 will tend to p_0 and p_2 will tend to p_{at}, so $\bar{p}$ will tend to $(p_0 + p_{at})/2$. The initial force in the spring will then tend to:

$$F_i = (p_{set} - p_{at})A_b + [(p_{set} + p_{at})/2 - p_{at}]A_{sb} \quad (8)$$

The excess force is somewhat similar to the clamping force discussed by Pearson[4], and has an important bearing on the variation in over-pressure.

To complete the set of equations, it should be noted that the flow areas for passages 1 and 2 are related to lift as follows:

$$A_1 = \pi d_b x \quad \text{and} \quad A_2 = \pi d_h (x + k) \quad (9)$$

Also, discharge coefficients for the passages and nozzle have to be applied, to account for departures from ideality.

Method of solution

The sets of equations for the two passages and nozzle described above can be solved iteratively for a given valve geometry, if p_0, p_{set} and p_{at} are specified, and if a valve lift is assumed. A Fortran 77 program was developed to do this, by starting with assumed values of the various intermediate pressures and iterating until the mass flows through the flow passages became equal. Eqs (6) and (7) were then used to calculate the magnitude and direction of the net force on the valve disc, and a new lift estimated accordingly. Successive repetitions produced the value of lift giving force equilibrium.

This technique was used to find the lift versus p_0 characteristic. In order to examine the conditions under which the valve first lifted, it was reasoned that, as p_0 increased towards the set pressure and the downward force on the disc (and hence on the valve seat) progressively reduced, a very small flow became possible, accounted for in the program by introducing a very small value of lift. The value of p_0 at which the net force on the disc changed from acting downwards to upwards was thus the over-pressure, and thereafter an equilibrium lift position was computed as already described. In this way a complete pressure versus lift characteristic was obtained from valve-lift to snap-shut.

Results from the computer program

A set of characteristics predicted by the program for a pop valve working with air and with a set pressure of 5.0 bar is shown in Fig 4. The valve parameters used were based on those typical of a 12.5 mm valve for an air receiver. The graphs may be interpreted as follows.

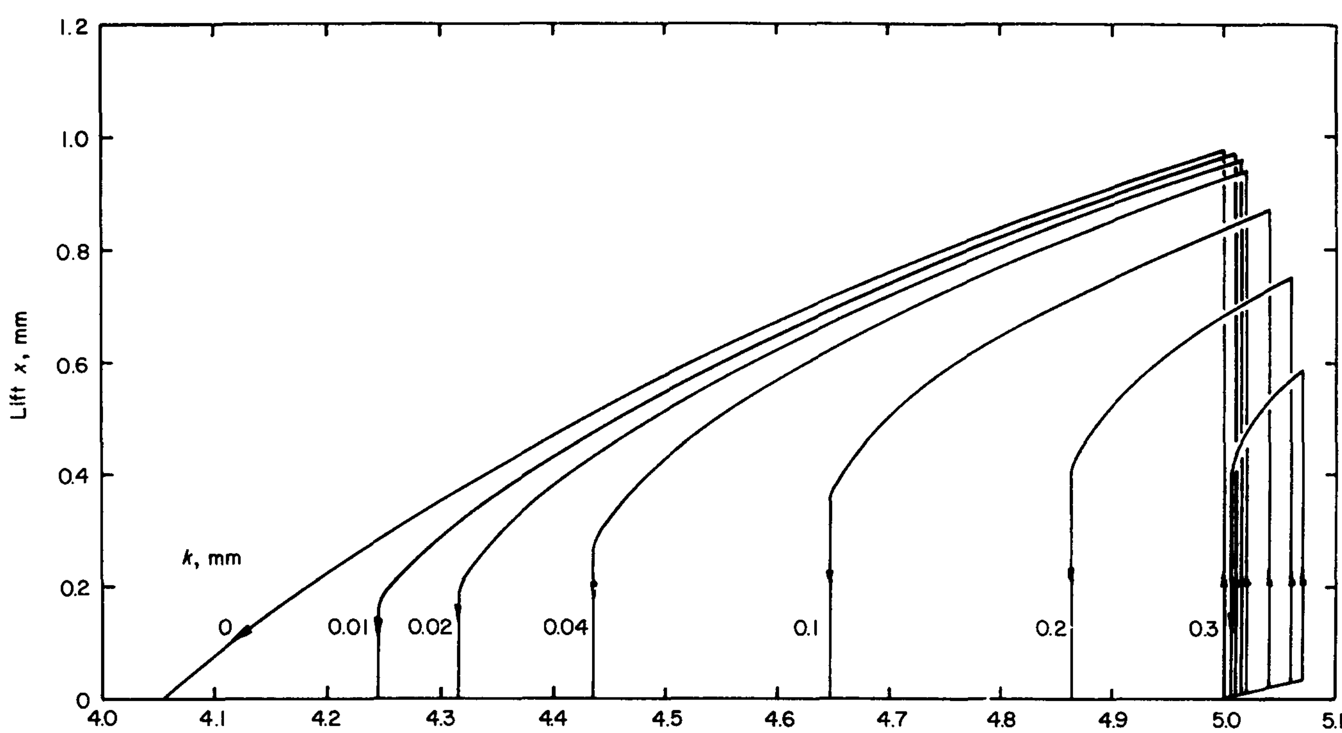


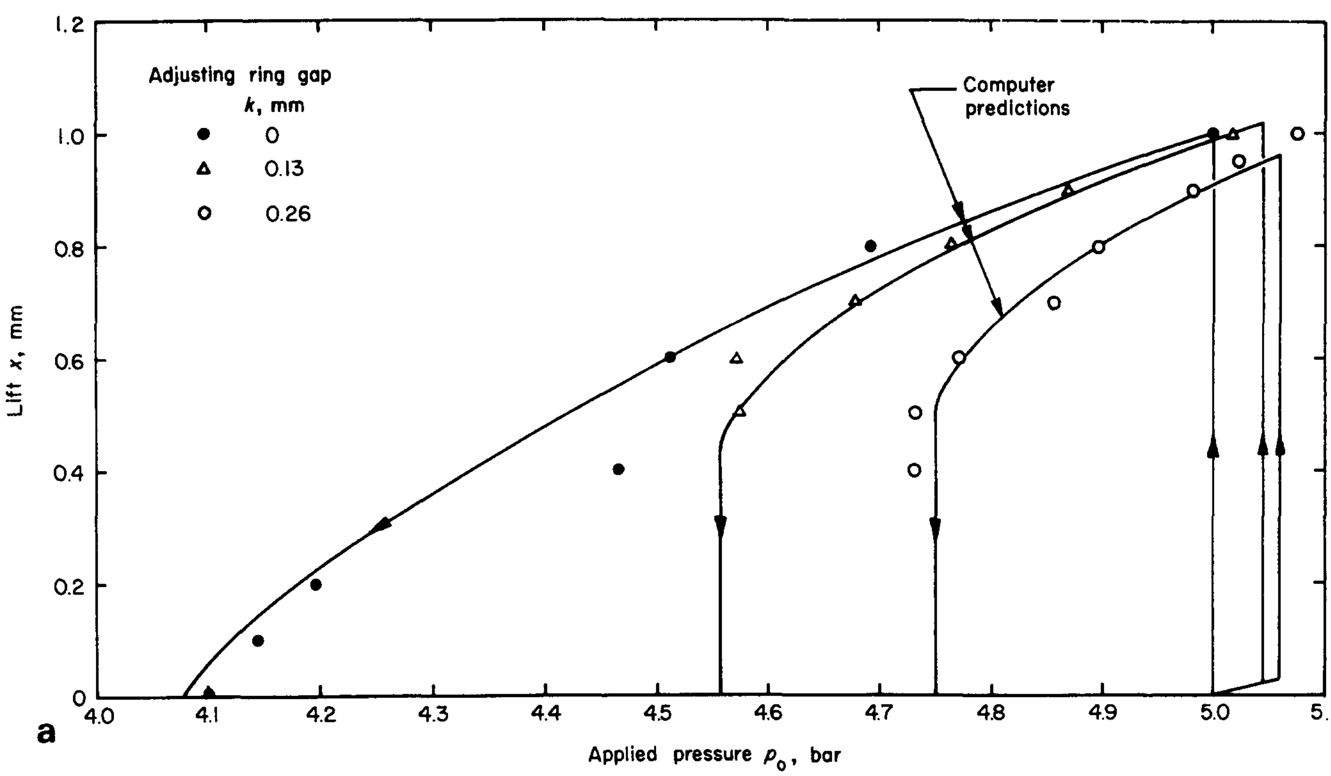
Fig 4 Comp7ter model prediction of Pop valve pressure versus lift characteristics for various settings of the adjusting ring gap. The set pressure is 5.0 bar (abs)

Considering the condition when $k=0$ (ie the adjusting ring is just touching its seat when the valve disc is on its seat), as the pressure beneath the valve rises to equal the set pressure, the valve snaps open to a maximum lift (in this example $x=0.98$ mm). This occurs because the pressures in the huddling chamber and body are such that the upward forces always exceed the downward forces until equilibrium is reached at maximum lift. Assuming that the discharge rate then exceeds the supply rate, the pressure will begin to fall, and the lift decreases steadily with pressure until the valve is shut, when the pressure in this case is 4.05 bar, ie 81% of the set pressure (or the blowdown is 0.95 bar). There is no snap shut action, and in practice the valve would probably continue to 'feather'.

It can be seen, however, that even small values of k influence significantly the action of the valve. Consider the characteristic for $k=0.04$ mm. The valve does not snap open until a small over-pressure has been reached, in this case 0.02 bar. The maximum lift is reduced slightly (0.94 mm), but the most significant change is that as the applied pressure reduces there comes a stage where the valve is unstable and snaps shut. The pressure for this example is 4.43 bar, ie a blowdown of 0.57 bar. A check on the force balance shows that in the stable lift zone, the pressures due to the flow are such that an equilibrium position can be achieved. However, once the lower limit of the stable zone is reached, the downward forces always exceed the upward ones, so the valve drives itself tightly shut.

Further increase in k produces a reduction in blowdown (which is desirable), but also produces an increase in the over-pressure and a decrease in the maximum lift.

For pressures between the set pressure and over-pressure there are, in fact, two attainable solutions to the set of equations, one with small lift and one with high lift. Since the valve is moving away from its seat, the small lift solution is reached first. As overpressure increases, the



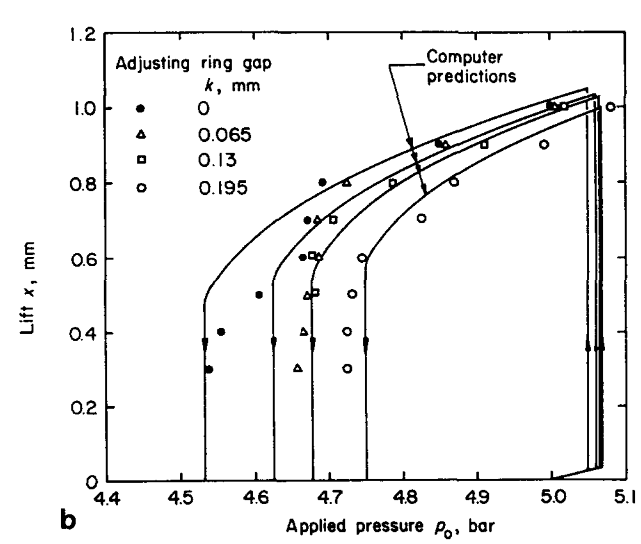


Fig 5 Comparison between computer predictions and experimental data from a $\frac{1}{2}''$ (12.5 mm) Pop valve. The set pressure was 5.0 bar (abs) (a) with solid valve disc (b) with 3×2.38 mm diameter holes in the disc, connecting the huddling chamber to the space above the disc (this was the valve's normal configuration)

upward forces increase, increasing the lift, and a point is reached at which the upward forces again exceed the downward ones so the valve rises rapidly to the high lift solution. As the pressure beneath the valve now decreases, the high lift solution to the equations is followed until the condition is reached, as before, when the downward forces exceed the upward ones and the valve snaps shut. As k increases, the difference between the small and high lift solutions diminishes, ie the snap-open action becomes less pronounced.

The optimum setting for k clearly is a compromise between the conflicting effects on blowdown, over-pressure and maximum lift. For the example given here, a value for k in the region of 0.1 mm would probably be acceptable.

Comparison with experimental data

Tests were made using a Bailey $\frac{1}{2}''$ (12.5 mm) Air Pop Valve, nominally rated at 6.0 bar absolute working pressure. The valve was mounted on an air receiver and this was supplied with air from the laboratory air-line through a control valve. Because of restrictions on the pressure available, the valve was adjusted for a set pressure of 5.0 bar absolute. The pressure was recorded by a transducer mounted below the valve, and the lift was estimated from movement of the valve stem, using an electrical contact device on a micrometer adjustment. Both readings were displayed on a 2-pen chart recorder. The valve parameters were:

Valve bore diameter	$d_b = 12.5$ mm
Huddling chamber diameter	$d_h = 22.9$ mm
Valve seat width	$s_b = 0.5$ mm
Adjusting ring seat width	$s_h = 1.11$ mm
Spring stiffness	$S = 10$ N/mm

This valve, in common with many others, had holes drilled from the huddling chamber to the top of the disc. This amounts to having another nozzle in parallel with the flow past the adjusting ring and can be taken into account with reasonable approximation by adding a constant ($= Nd_N^2/4d_h$) to the value of k, where N is the number of holes of diameter d_N.

The results for different settings of the adjusting ring are shown in Fig 5. To compare the predictions from the computer model with these results, assumptions had to be made about the various discharge coefficients, and the curves presented used values to give the best fit to the experimental data. It was expected that the valve conditions would have little influence on the discharge coefficients for passage 1 and the nozzle, and this was confirmed in that good fits with the data were obtained with values for the coefficients only varying in the range 0.64 to 0.70 and 0.90 to 0.94, respectively. Passage 2 was influenced by the adjusting ring setting and by the blocked or unblocked holes, and the discharge coefficient was found to require a wider range of 0.76 to 1.00.

Conclusion

The analytical model described here gives satisfactory prediction of pop valve pressure versus lift characteristics. It has been shown that the snap action of the valve arises from instability between two possible solutions to the set of equations, between certain applied pressure limits. The known sensitive effect of the adjusting ring setting on the characteristics has also been clearly confirmed. The finite widths of both the valve seat and adjusting ring seat and the associated assumed small leakage flow when the applied pressure approaches the set pressure have significant effects on determining the lifting conditions and hence over-pressure.

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