Simplified Theory Applied to Automatic Compressor Valves

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1.1. PURPOSE AND SCOPE OF THE PAPER.

The most important parts of any reciprocating machine, which require the greatest skill in design, are the devices which control the inlet and outlet of the working fluid. We will treat here one group thereof, namely the most frequent types of automatic valves for gas compressors; not included are valves for liquid pumps or positively actuated valves, such as the poppet valves of internal combustion engines or the slide valves of steam engines, although these latter are used on some compressors, particularly vacuum pumps.

Great effort has been spent on the construction, design, and computation of these valves nevertheless much confusion still seems to prevail, leading to misunderstandings between valve designer, compressor manufacturer and operator.

Some prior publications [1, 2] go to great lengths to determine the exact motion of the valve during opening and closing, which in my opinion is not worth the trouble in the first place, and secondly they arrive at equations much too unwieldy for practical use - even with computer aid. This complicated, though admittedly basic, approach and its display of high brow mathematics is the less justified as none of the equations could be arrived at without several simplifications, such as infinitely long connecting rod, zero clearance, exactly adiabatic process, constant discharge coefficient, and the stipulation that the smallest passage area is through the lift; all of which are not correct, some of them, indeed, very far from the truth, and the last one, of course, is aimed at by every designer, but often unattainable. By the way, some of the profound conclusions from these painstaking labors are either plainly obvious or, worse yet, misleading, one of the regrettable side effects of the modern trend towards "engineering science" with its mathematical fireworks as opposed to "engineering" with its eye on the hardware. E.g., it was brought out that the most important parameter in valve design is a coefficient "const * V / A " with the constant depending on the gas properties; this boils down to the unsurprising statement that the flow velocity through the valve is the determining factor. Or it was stated that, all other things being equal, the volumetric efficiency improves with increasing area of the intake valve; now, increasing the valve and keep everything else constant would be an extremely nice trick, if one could do it; alas, it is one of the basic facts in the valve designer's life that one has to pay for increased area by increased clearance space and its devastating effect on volumetric efficiency.

1.2. GENERAL APPROACH TO THE PROBLEM.

We proceed to determine the characteristics of each valve (most of them not dimensionless numbers), regardless of the compressor, in which it may be installed, and regardless of the fluid, for which it may be used; then we determine what properties are required for a particular application, and match them up. Very often valves are nowadays not manufactured by the compressor builder, but by companies specialized in this field, and it is obviously desirable to use the same type of valve, no matter by whom built, in many different compressors, for different pressures, different gases, and perhaps different speeds; often it is also desired to turn a valve around and use it either for intake or discharge. Where the use of several sizes is unavoidable, it will be advantageous to use the same components, such as plates or springs, in various combinations. E.g., the biggest valve of a series may have four nested ring plates, the next smaller size uses only the three inner ones, and so on; variation of specific load may be obtained by applying the same spring to plates of different sizes etc. Sticking to standard pitch circles not only reduces the number of jigs and fixtures, but also simplifies the design work, because we need the data for each component separately anyhow, before we combine them thus, we need only keep our records reasonably organized, and can then easily select the suitable combinations. Special
design for unusual cases is very seldom necessary, and in the overwhelming majority of cases standardized -- even "off-the-shelf" valves are not only economically desirable, but can be used without loss of efficiency.

Let me now demonstrate that the proposed simplifications are permissible and justified.

2. GAS VELOCITY THROUGH THE VALVE.

The most important consideration is the pressure drop through the open valve

\[ \Delta p = \frac{2\pi \cdot \mu}{v^2} \cdot 144 \cdot 3600 \]

The gas density at the intake can almost always be calculated by the perfect gas law

\[ \rho = \frac{P_1 \cdot \mu}{10.72 \cdot T_1} \]

and at the discharge by

\[ \rho_D = \rho \cdot \varepsilon \]

For very high pressures or vapors, of course, consult the usual tables.

\( v \) is the velocity through the narrowest area, also called "effective area". An old rule of thumb was that it should not exceed 6000 fpm, but in modern high-speed compressors this limit cannot be observed any more, and twice as much, or even more, must often be put up with. I shall in the following use the "mean" velocity, defined by

\[ v = \frac{\pi}{4} \cdot D_5 \cdot N_1 \cdot A \cdot 1/12 = 238 \text{ K/A} \]

which in other words the velocity that would prevail, if the flow were constant over the entire stroke. Actually the flow depends, among others, on the momentary piston speed, which varies between a maximum and zero. A kinematic analysis -- which can for lack of space not be reproduced here -- shows that the error through basing our computation on the mean piston speed

\[ c = \frac{N_1}{\delta} \]

is permissible for our purposes. Also the area varies from zero to the full opening \( \delta \). However, in a well designed valve the time consumed in the opening and closing motions should only be small in proportion to the entire open-time. We are interested in the pressure drop only to compute the extra power necessary to overcome it, the shaded area in Fig. 1. For this purpose it is completely satisfactory to know the average drop \( \Delta \), and there is no need to waste time and effort in painstakingly investigations the law of motion, which after all only determines the shape of the little "humps" in the diagram, and which also depends on parameters outside the values, about which we can only make educated guesses.

For the flow coefficient \( \varepsilon \), we assume for a starter a constant value of 0.575; this is only a rather rough approximation, but not too far from the truth. We arrive, thus, at

\[ \Delta_0 = \left( \frac{\pi \cdot \mu}{v^2} \right) \rho \]

Without any calculation we know that we want to keep the velocity as low as possible; thus, we strive to pack as much passage area into a given port hole as we can without incurring some other disadvantage, such as excessive clearance volume or high cost. We want the pressure difference to open the valve and to keep it open, as low as possible, and on the other hand the spring load high enough to produce a rapid closing motion. As in all engineering problems these contradicting demands must be compromised. In establishing performance limits we must rely on comparisons with other valves or other operating conditions, and the errors caused by our simplifications will be essentially alike in almost all practical cases, so our comparison remains valid.

3. LOCATION OF THE VALVES.

The valves may be installed either (a) in the cylinder barrel, or (b) in the head, or (c) intake valves in the piston. (a) usually results in larger clearances, (b) limits the space available, particularly on the crank end of double-acting machines, and requires removal of pipelines for repairs, (c) makes large area available with small clearances, lets the inertia assist the valve motion and permits uni-directional flow, reducing the irreversible heat exchange, but requires pressurizing the crank case, may cause oil carry-over and prohibits installation of automatic control devices in the intake valves.

4. TYPES OF VALVES.

We consider first valves with rigid sealing elements and separate springs, as distinguished from those with flexible seals. The "plates" of the first group move generally at 90° to the seat, whereas the "blades" of the second group turn, in part or in whole, relative to the seat.

The prototype and most frequent construction of the first group are circular ring plate valves with separate helical springs, which we shall treat; however, for such variations as straight plates and/or leaf springs (Ingersoll-Rand's Channel valve, Sullivan's Cushion Valve) or slotted plates with elastic guides (Kogler, Hoerbiger), or even old fashioned poppet valves, or rectangular valves (which can be, and have been built) most of our results remain valid, or the required modifications will be obvious.

The second group of valves comprises those constructions where the lid itself is flexible and provides its own elastic force, the so-called reed valves, where the strips are fastened to the seat on one end (Burn) or the so-called Feather Valves (Worthington), which have loosely installed bending strips.

5. RIGID SEALING ELEMENTS.

The two characteristic positions are, closed, when the plate is in contact with the seat, and fully open, when the plate is far-
The area of the pressure difference on the open or closed valve is, see fig. 2, 
(24) \( A_0 = \pi D^2 \) ; (25) \( A_C = \pi D^2 \left( B_C + B_A \right) / 2 \)

The closed plate is subject to the full static pressure difference, and will begin to open as soon as this overcomes the spring load; on the other hand, to hold the valve open the dynamic pressure drop, created by, and necessary for, the flow through the 
(26) \( \Delta p = \frac{p_C}{A_C} \) and (27) \( \Delta p = \frac{p}{A_0} \)

\( \Delta p \) is easily determined from the geometry and \( \Delta p \) can be found from Eq. (29). Now always \( p_0 > p \) and \( A_C > A_0 \), but the ratios \( p_0 / p \) and \( A_C / A_0 \) determine whether \( \Delta p \) or \( \Delta c \) is greater. \( \Delta c \) should be made \( \Delta p \) to keep the valve sufficiently open, however valves with opposite have given satisfactory service. A high ratio \( p_0 / p \) is allowed, assuring that \( \Delta c / \Delta p < 1 \) is also desirable for consideration of spring stress; unfortunately for given \( p \) and maximum stress this can only be achieved by a great number of turns, requiring deep grooves and leading to large clearance on the intake valves; so, again we must compromise.

Opening motion: We need not worry, opening will be quick enough in all practical cases, because then the piston moves rather fast, on the closed or slightly open valve pressure will build up and more and more force becomes available for plate acceleration. A slower than desired opening would only in crease somewhat the "hump" in the diagram. On the other hand, high velocity means heavy impact on the guard; we can soften the blow by heavy springs, but Eq. (27) imposes a limit on the permissible spring force, and all we can do, is make the springs as heavy as this condition allows.

Closing motion: Ideally we want the valves, intake, as well as discharge, to close when the piston arrives at the dead center. Before this the piston moves slowly, the flow and, thus, the pressure drop are reduced, but in the valves, closed practically during the piston reverses, back flow would set in, although this would tend to close the valve, it would also tend to equalize the pressure, and so the valve would act rather sluggish and allow quite a bit of back flow with its dreadful effect on volumetric efficiency.

Anyway, we do not want to wait for this back flow, but want to close the valve before it develops, which means it must close by the spring force alone within a very short time; if \( F \) gets a little help from the pressure, so much the better. We do not care for the law of motion, but simply want to limit the time of travel through the entire lift.

The acceleration of each plate is, of course, simply the spring force divided by the accelerated mass
(30) \( a = F / m \)

It is most practical to express the spring forces in "g"s, viz.
(31) \( \frac{F}{m} = g \cdot \frac{w}{W} \)

If the acceleration were constant over the entire lift the time for the closing under the influence of the spring(s) alone is
(32) \( t_c = 1000 \sqrt{2L / (12a)} = \sqrt{5200 \cdot L / g} \) ms

The impact velocity, with which the plate hits the seat becomes
(33) \( v = 60 \sqrt{2L \cdot a / 12} = \sqrt{19300 \cdot L} \) ft/min

The angle through which the crank moves during the time of closing is
(34) \( \phi = t_c \cdot \sqrt{6} / 180 \)

Substituting from Eq. (32) we get the simple and most useful relation
(35) \( \phi = \sqrt{2 \cdot L \cdot g / W} / 2.31 \)

We can thus stipulate either directly the angle, through which we will allow the crank to move during the closing time, or find it on one of the well known tables for a stipulated position.

Actually the acceleration decreases as the spring extends and its force diminishes, but the error is negligible; proof omitted for lack of space.

Valve Characteristics: The following parameters should be computed as yardsticks for the quality of design and guides for future estimations: Areas through seat, guard and lift \( A_3, A_0, A_L \) and effective area, which is the smallest of the three and ought to be \( A_L \), clearance spaces in seat (discharge) and guard (intake) \( Z_2 \) and \( Z_0 \), closing time \( t_c \), impact velocity \( v \), opening and holding pressures \( \Delta p \) and \( \Delta c \) of acceleration and spring stress and the following indicators of efficiency: area efficiency \( \eta = A / A_0 \) (0.25 about average, 0.30 about tops), clearance factors \( \eta_0 \) and \( \eta_5 \), \( Z_2 / \eta_5 \) and \( Z_0 / \eta_0 \) (exact \( \xi \) from fig. 10), 2", low values for slow speed and low pressures, i.e. high lift and weak
springs, allowing shallow grooves, expect $\xi_s$ from $\frac{3}{4}$" to $\frac{1}{3}$", lower values for low speed and small pressure differences, higher values for high pressures, requiring thick seats for strength, $\gamma$ (for air at atmospheric intake abt. 7 to 15, for discharge, high speed and/or heavy gases 40 up to 120); impact velocity 200 to 300 acceptable for steel, $\beta_s$ should be over 0.5 for fatigue.

Design Procedure for existing compressor:

Frequently valves have to be designed to fit into existing port-holes; in all probability they will be smaller than the valve designer likes, but he has no choice than to make the best of it, unless he foresees too much trouble and refuses the job. Except for the first step the procedure is the same as described in the following.

Design for a new compressor: To get things started we first assume a mean gas velocity of 6000 fps; then the total effective area "per corner" (valves per corner are the number of valves of the same kind on each working end of each cylinder), must be

\[ A = \frac{7}{4} \times 6000 \times 12 = 46000 \]

Assuming an obtainable $\eta$ we find number and size of the desired port-holes; if they can not be accommodated, we must compromise. The port-hole diameter limits the diameter of the outermost ring plate, from which we go towards the center, laying out the inner rings. Here the first difficulty arises: because the travel through the lift must be completed very fast and on the other hand the acceleration is limited (too heavy springs would cause too much pressure loss) the lift is limited; obviously the less time we have, the lower the lift must be. The best utilization of our resources is to make all passage areas $A_a$ and $A_r$ alike; this means $3a_d = 2L$ (Fig 3). Now it is very difficult and almost impossible to make slots in the seat narrower than $\frac{3}{16}$", thus we would like a minimum lift of $\frac{3}{32}$"; furthermore a certain overhang $b_c$ (abt. 0.013 to 0.020) and contact $b$ (0.050 to 0.090) are required for a reliable seal. Consequently, if we reduce the lift, we can not narrow down all the $b$'s below a certain minimum, in order to accommodate more rings, or in other words with low lift our area efficiency $\eta$ decreases. This is the main dilemma, between the horns of which the valve designer is caught, viz. for a compressor with given bore and stroke we ought to provide more valve area the faster it runs, but actually we must reduce the lift and have less and less available. Nevertheless, whether the valve designer likes it or not, the trend is towards higher and higher speed for obvious reasons of reducing weight, space and cost of the machines.

Strength: The thickness of plates and seat is determined by strength, that of the guard by the depth of the spring grooves. Details must be omitted here, suffice it to say that in most cases deflection is determining rather than ultimate strength. E.g. if the admissible fatigue stress of the material $\sigma < 1000 \sqrt{\frac{\text{max}}{b}}$, the plate need only be calculated for bending stress, otherwise (i.e. for low pressure) for deflection. If the ratio of the height of the ribs of the seat to the slot width $H_s/d_s > 2$ the ribs may be considered constrained beams, if the ratio $< 1$, they must be considered freely supported at the ends.

In order to get shallow grooves rectangular spring wire is preferable; if the sides of the wire cross-section have a ratio $m = b$, wherein $b$ is always the greater regardless of the orientation relative to the spring axis, we can introduce an "equivalent diameter for spring constant"

\[ d^* = b \cdot 1.13 \frac{m^*}{m 
\]

and an "equivalent diameter for stress"

\[ d^0 = b \cdot 1.04 \frac{\sqrt{m^0}}{m} \]

and then use one of the convenient special slide rules available for round wire [5].

Clearance. Refer to Fig.3. The clearance space attributable to the valves consists of 1) $V_g$ the gas filled volume within the valve itself (in the guard for intake, seat for discharge), which can be found by elementary geometry, a tedious job, but simple and easy to computerize; 2) the valve must be set back in the port-hole by a small safety margin $M$ of abt. 0.060 to 0.090", creating a clearance $Z = M + A$; 3) if installed in the cylinder barrel, the intersection of port-hole and cylinder "c" $Z_c = D_p/(20D_p)$ [6]; 4) if the cylinder head overlaps the port-hole, access may have to be provided by a throat $b$.

\[ A* = \frac{D_p^2}{2} \left( \gamma + \frac{1}{2} \right) - \frac{D_p^2 \cdot D_p^2}{4} + \frac{\gamma}{2} \]

\[ y = D_p - \sqrt{\frac{D_p^2 - 25}{4}} \]

\[ A^0 = 2 \arcsin \left( \frac{D_p}{D_p} \right) \]

\[ A^* > A^0 \] if the throat $A^0$ for intake, no throat is necessary, otherwise the wedge-shaped volume can be estimated as

\[ V_c \approx (A^0 - A^*) \cdot e/3 \]

For discharge replace $A^0$ by $A^*$. The relative clearance

\[ z = \frac{Z_c}{A^0 \cdot S} \]

should in a well designed valve be kept at abt. 0.05 for slow compressors, while for fast ones 0.10 may be necessary, and for very fast ones 0.15 may have to be put up with.

A more systematic approach, eliminating most of the trial and error has been worked out, but cannot be presented here for lack of space. Also the special case of valves installed in the piston and subject to inertia must be omitted.

5. FLEXIBLE SEALING ELEMENTS, END CONstrained

The sealing element of a Reed Valve, Fig.4, is a strip, most often steel, clamped tight to the seat on one end, bending away on the
other under the influence of pressure, thus providing a wedge-shaped passage way. Most often the blades are rectangular and flat in the free, unstressed condition, held in the sealing position only by the excess of pressure on the side away from the seat, for simplicity called "top side". The elastic force is zero in this position and the slightest force from the underside will begin to open the valve, or 

$$P_c = 0 \text{ and } \Delta = 0$$

The blade is deflected by an equally distributed load, see Fig 4.

From the equation of the elastic line of a cantilever beam we obtain

$$(85a) \ L = \left(\frac{8\cdot10^3}{(8\cdot10^5)}\right) \sqrt{\frac{J}{E}}$$

$$(85b) \ L = \left(\frac{3\cdot10^3}{(2\cdot10^5)}\right) \sqrt{\frac{J}{E}}$$

$$F$$

by integrating we find the wedge-shaped passage area on one side

$$(87) \ A = \frac{F}{L} \cdot 1$$

$$= \frac{F}{L} \cdot \frac{L}{2}$$

$$= \frac{F}{2} \cdot \frac{L}{2}$$

$$= \frac{F\cdot L}{2}$$

$$\Delta \approx 0.85$$ in most cases. If the point of constraint does not coincide with the end of the slot, small corrections may be taken from a table, which space does not allow to reproduce here.

For a blade, free to deflect, the bending moment is at the fully open valve

$$(93) \ \Delta = 0 \ F/2 = L^3 \cdot H^3 \cdot F/2$$

with $J$ a section modulus of $3 \cdot 10^3 / 6$.

$$(94) \ \Delta = 0 \max = 3 \cdot 10^3 \ F^2 / 6$$

For the closed blade with a modulus $F^2 / 6$

$$(97) \ \Delta = \frac{3 \cdot 10^3 \ F^2}{6}$$

$$= \frac{3 \cdot 10^3 \ F^2}{6}$$

The stress on the open valve is the greater.

Where construction limits the lift to a lower value, we do not fully utilize our material, and have to use the actual lift and stress in our equations, rather than the permissible maxima.

Combining (88), (1b) and (29) we obtain

$$(99) \ \Delta = 0 \ k^2 \cdot c / (400 \cdot F^2 \cdot L^3 \cdot A^2 \cdot Z^2)$$

equating this with the value from (85b)

$$(100) \ L = \frac{3 \cdot 10^3 \ F^2}{L} (6.44 \ H / \sqrt{A^2 \cdot Z^2 \cdot C})$$

Substituting $L$ from (85b) into (99) we get

$$(101) \ \Delta = 0.104 \cdot \frac{H^2}{L^3} \cdot \frac{3 \cdot 10^3 \ F^2}{(\sqrt{A^2 \cdot Z^2 \cdot C})}$$

For instance for the intake valves of an air compressor with $A = 0.95$, $Z = 0.575$, $F = 0.78$ in steel, $29.5 \cdot 10^3$ we get

$$L = \frac{3 \cdot 10^3 \ F^2}{L} / (2890 \ H)$$

$$\Delta = 0.104 \cdot \frac{H^2}{L^3} \cdot \frac{3 \cdot 10^3 \ F^2}{(\sqrt{A^2 \cdot Z^2 \cdot C})}$$

Sometimes the "dynamic pressure height" is convenient for computation, viz.

$$(99a) \ \Delta = \frac{0.104 \cdot F^2}{L^3} / \sqrt{A^2 \cdot Z^2 \cdot C}$$

The bending moment that restores the blade to its unstressed shape is proportional to the deflection, which is the necessary and sufficient condition for harmonic motion (the blade will not vibrate in a higher mode, as parts cannot swing below the neutral line, viz. the valve seat). We can, thus, use Rayleigh's method to compute the closing time $J$, with following results

$$t_c = 78 \cdot [\sqrt{\omega / E}] \cdot F / H$$

Note that Eq. (109) does not contain the lift, the closing time does not depend on it, in analogy to a swinging mass (pendulum), whose period is independent of the amplitude.

6. OTHER FLEXIBLE SEALS.

Many different variations have been used and almost infinitely many more can be thought of. For all we can establish one relation between the opening $A$ and the load $P$ created by the pressure drop $\Delta$, using the laws of elasticity; and a second relation through the laws of fluid flow; in addition these two equations contain only the valve dimensions, physical constants of the fluid and the material, and the required flow, and they can, thus, be solved for the unknown quantities.

In the first place, the valves can be made trapezoidal and be constrained either on the wide or the narrow end. Theoretically, although at excessive manufacturing cost, the thickness could be varied, e.g. to form a beam of uniform stress. An analysis, that cannot be presented here, shows that our equations remain valid with different numerical constants.

Also the point of constraint can be removed from the end of the slot, and the blade be so formed as to have an initial tension. The Gutermuth Flap, Fig. 5, goes to the extreme. To speak, we find

$$\Delta = \frac{C \cdot E \cdot H^3}{(6 \cdot 10^5 \cdot F)}$$

wherein $C \cdot E$ is the developed length from the point of constraint to point $C$, $E$ can be found by solving an equation of the third degree, and then the other parameters Analysis of the so-called "Feather Valve" (Worthington) with its freely supported blades shows that our equations can be used with changes only of the numerical factors, e.g. in Eq. (85) $A$ is to be replaced by $76.8$ and $3 / 2$ in Eq. (85b) by $3 / 32$.

7. MULTI-ELEMENT VALVES.

If a ring plate valve contains several rings, try to make the t's alike, that is with equal lift (as practically always the case) chose the springs for equal $P$, rather than for equal $\Delta$. With blades of different length in one valve (e.g. to better fill out circular space) we proceed in principle as before, but have to re-write our equations in terms of area rather than lift, viz.

$$\Delta = 0 = k^2 \cdot F^2 / (400 \cdot F^2 \cdot L^3 \cdot A^2 \cdot Z^2)$$

$$= \frac{0.104 \cdot C \cdot E \cdot H^3}{(6 \cdot 10^5 \cdot F)}$$

3. CONCLUSION.

I have tried to establish yardsticks for performance. Other factors influencing the choice - tradition, existing tooling, salesmanship, and predilection for one's own brainchild - are beyond the scope of this paper.
9. SYMBOLS.

| A | Area (see Note) | sq.in. |
| B | Width | inch |
| C | Piston Speed | fpm |
| D | Diameter (see Note) | inch |
| E | Modulus of Elasticity | psi |
| F | Length of Reed | inch |
| G | Thickness of Plate (blades) | fpm |
| H | Impact Velocity | cfm |
| I | Diameter (see Note) | inch |
| J | Margin (Bending) | inch |
| K | Speed | lb/min or strokes/min |
| L | Load | lb |
| M | Stroke | inch |
| N | Temperature absolute | deg |
| O | Volume | cu.in. |
| P | Weight | lb |
| Q | Clearance Volume | cu.in. |
| R | Developed Length | inch |
| S | Width of Spring Wire | inch |
| T | Diameter of Spring Wire | inch |
| U | Gravity Constant | deg |
| V | Height of Spring Wire | inch |
| W | Number of Valves per Cylinder | -- |
| X | Shape Ratio of Rectangular Wire | -- |
| Y | Polytopic Exponent | psi |
| Z | Dynamic Pressure Height, ft of fluid | psi |
| 0 | Time | ms |
| P | Gas Velocity (see Note) | ft/sec |
| Q | Density of Solid Material | lbs/cu.in. |
| R | Relative Clearance | -- |
| S | Acceleration Factor | -- |
| T | Pressure Ratio | -- |
| U | Flow Coefficient | -- |
| V | Area Efficiency, A/AH | -- |
| W | Molecular Weight | -- |
| X | Clearance Factor | -- |
| Y | Density of Fluid | lbs/cuft |
| Z | Stress | psi |
| 0 | Angle of Twist | deg |
| P | Central Angle | deg |
| Q | Difference (see Note) | -- |
| R | Area Coefficient | -- |
| S | Spring Ratio | -- |

Subscripts

- a average
- b closing
- c inner, initial
- d discharge
- e guard
- f intake
- g margin
- h gap
- i combined
- j effective
- k open, outer
- l overhang
- m port-hole
- n piston
- o seat, slot

Without subscript symbols mean the following:

- L Valve Lift
- v Highest Gas Velocity
- W Effective Area
- D Pitch Diameter
- B Width and W Weight of Plate or Blades
- A without qualifier = Pressure Drop

10. REFERENCES.

5. Available from Associated Spring Corp., Bristol, Conn.