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TECHNOLOGY IN SEVERE SERVICE CONTROL VALVES

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Summary

Severe service control valves are critical for safe, reliable and efficient operation of power plants. Such critical applications must be looked at differently from general service control valves because these applications have their own specific set of requirements for good long-term performance. The performance limits of control valves in such services is clearly a function of the technologies in them. Discussion on severe service valves is presented in order to aid the application of correct technology in such critical services.

Introduction

Control function is a feature common to all control valves. This is pretty much where the commonality ends. While there may be over a thousand control valve applications in a typical nuclear plant, those critical for safe and reliable operation are much fewer. In addition to these applications, there are other applications where valve performance is critical for efficiency (MWe) of operation. A majority of these valve applications come under the category of severe service valves. They are much fewer in comparison with the total valve population, however they are the ones that account for a majority of the valve issues.

Standards for control valves have evolved, as in other industries, over time. The primary object of these standards has been to ensure safety of the public and of the equipment, consistency in sizing for flow capacity and for ease of installation. The standards generally dictate minimum design practices for safe operation under all the known operating conditions, with sufficient margin of safety. Typically, this covers pressure-boundary integrity and mechanical stresses, considering both static and dynamic conditions.

The standards for control valves do not provide clear performance guidelines, nor do they provide any design rules for minimizing potential for problems in a given application. This makes selection of control valves for severe service applications all the more important. There is certainly an abundance of data in this area - both, in terms of solving problems, and in terms of having experienced the problems.

The key in solving severe service valve problems is the technology. Because of the nature of such applications, it becomes important to recognize the effects of off-design conditions. While valves in general service category are often available off-the-shelf, such an approach for severe service applications is risky.

Recognizing Severe Service Applications

There is sufficient operational experience that can be pooled together to recognize severe service valves. Some of the guidelines for doing so are given below.

1. User experience - If there is a continuing operational, maintainence or reliability problem with control valves in a given application.

2. Application experience - Valves in many applications in nuclear plants are known to experience severe service conditions. Some of the applications in nuclear plants are:

    - Pump minimum flow recirculation
    - FW Control valves
    - Turbine Bypass (atmospheric dumps or condenser dumps)
    - Emergency Heater Drain Level control
    - RCIC, HPCI, RWCU, Core Spray
    - CVCS Letdown
    - HX Bypass
    - SG Blowdown
    - Service Water Flow control

3. High Vibration

4. High Noise

5. Poor process control at design or off-design conditions (MWe improvement issue)

6. Leakage (MWe improvement issue)

7. Mechanical damage to valve components

8. Damage to the system around the valve

9. Environmental Qualification - High reliability requirement in harsh environments for a control valve application can also be treated as a category of severe service valves.

Eliminating Problems in Severe Service Applications

Having established these guidelines for recognizing severe service applications, one can then turn to the requirements that valves must meet in order to provide good long-term performace.

The traditional approaches to addressing such problems have been by continual maintainence or brute force or both. The brute force approach covers solutions such as using harder materials where there is erosion, covering up with insulation to reduce noise, more supports to reduce vibrations, and so on ... and so forth. These merely try to cover up the symptoms. They do not address the root cause at all. Sometimes just changing to another valve is tried without thoroughly evaluating if the solution being tried truly eliminates the root cause of the symptoms observed. This is risky, because the result could very well be no improvement to marginal improvement, and expensive, because of what it takes to change out valves in nuclear plants.

The most reliable way of eliminating problems is by addressing the root cause(s).

Identifying Root Cause of Problems

Identifying the root cause of the problems is the first step in ensuring that the technology in the valve is suitable for the specific application, whether in existing valves or in terms of potential in new application.

As stated earlier, all severe service applications are not exactly the same. This makes the details of the application requirements all the more important. However, most of the causes of problems in severe service applications fall in five categories :

- excessive fluid energy along the flowpath
- inadequacies in actuation system
- design details
- valve-piping interface
- poor maintainence/incorrect calibration

The last issue of maintainence and calibration is not a valve technology related issue. No amount of maitainence or calibration will offset the shortfall in performance because of the first four items indicated above. Most importantly, it should not be used, or allowed to be used, as an excuse for continuing valve problems.

Fluid Kinetic Energy Control

The first critical aspect of eliminating the problems is the control of fluid kinetic energy in the valve all along the flowpath. This has been the key in solving many valve problems as demonstrated by Persad et. al. [Reference 1- EPRI Valve Technology Symposium, 1997] which documented elimination of problems in a turbine bypass application. Miller and Stratton [Reference 2 - ASME Paper FEDS 97-3464] provide a good discussion of different approaches, illustrated in Figure 1, for controlling fluid kinetic energy control in severe service applications. These different approaches, leading to different technologies, provide different levels of protection in severe service application. As a result, judicious evaluation as to whether the level of fluid kinetic energy control is adequate in a particular application is called for in such cases. Generic guidelines (Appendix I), which include quantitative and verifiable criteria, for severe service applications are available in such evaluation.

Reference 2 also provide examples of how fluid kinetic energy control principle and other principles were applied to solve problems in four (4) different services, namely :

Residual Heat Removal (RHR),
Feedwater control,
Core spray and
Condenser Steam Dump (CSDV).

The generic guidelines in Appendix I have evolved from experience in a large number of applications such as these. They set quantitative and verifiable criteria that can be used to qualify the technologies for good valve performance in service.

Actuation Requirements

Another important aspect for control valves is actuator selection. This has been a focus of major work in the nuclear AOV Users Group (AUG). Much effort has been directed towards ensuring that the valve operators are reliable and will act when called upon to do so.

In critical service applications, in addition to being adequate as far as operability for a control valve, actuator sizing and selection is important in four areas :

- tight shutoff capability,
- good controllability under normal operating conditions,
- good controllability during transients, and,
- environmental qualification

Operability only means that a valve can be modulated when required to do so. It does not ensure adequate thrust for shutoff or being able to control the process satisfactorily. Since guidelines for achieving the required tightness of shutoff are not covered by any standards, it is critical to spell out quantitative criteria in ensuring that the actuator has sufficient thrust. In principle, even small positive force is adequate for tight shutoff if the sealing surfaces are perfectly machined. This is not practical because the real world is far from perfect in this respect. Even then, in the valve industry, the range of variation in recommended thrust for tight shutoff is wide; for e.g., for ANSI class IV shutoff, the recommended shutoff thrust varies from 40 PLI, or pounds per lineal inch of seat circumeference, to 400 PLI. Experience shows that sufficient conservatism in shutoff thrust is necessary where tightness of shutoff is important. This ensures that the tightness of shutoff is maintained for long-term service, well beyond just the factory test. Again, generic guidelines are available to ensure that such conservatism is built into the sizing and selection.

Operability has another aspect, which is good controllability under normal operating conditions. It is often overlooked that control valves are the final control elements in process control. Even the most sophisticated digital control systems (DCS) can not make up for limitations in the control valve performance. Small hysterisis and dead band, quick response to small signal changes and stable dynamic response are all key parameters that define good performance. All, or some, of these are essential depending on the service; for example, feedwater control applications require all these characteristics.

Control valve actuators come in many varieties - pneumatic, electromechanical and electro-hydraulic. Ritz [Reference 4] provides a good comparison on different types of operators. Electro-mechanical and electro-hydraulic actuators provide excellent characteristics in terms of low hysterisis and deadband, and quick response to small signal changes; on the other hand, it comes at the expense of some other desirable features. Pneumatic actuators feature mature and proven technology, are simpler in terms of construction and maintainence, and are most common in nuclear plants. However, the resolution of the control valve must be checked where fine controllability is required. This has been discussed by Leimkeuhler and Sherikar [Reference 3- EPRI Valve Technology Symposium, 1997]. Briefly stated, resolution, or the smallest change in control valve position, can be determined by,

$$\Delta X = (F_s - F_d)/K_p, \qquad \text{... Equation (1)}$$

where,

F_s is the summation of static friction forces acting on the plug and stem,
F_d is the summation of dynamic friction forces acting on the plug and stem, and,
K_p is the pneumatic stiffness of the actuator at valve lift corresponding to X.

It is clear from this relationship that higher actuator "pneumatic stiffness" means a smaller step is possible in response to corresponding signal change, hence better control.

Referring to the nomenclature in Figure 2,

$$K_p = 1.4*A*\{P_1/L_1 + P_2/L_2\} + K_{spring}$$ ... for piston actuators ...Equation (2a)

$$K_p = 1.4*A*P_1/L_1 + K_{spring}$$ ... for diaphragm actuators ...Equation (2b)

Typical actuator stiffnesses for piston actuators and diaphragm actuators are shown in Figure 3.

The step-change in flow-capacity corresponding to the smallest change in position that the actuator can accomplish can be determined from the slope ($\delta Cv/\delta x$) at lift X of valve flow capacity (Cv) versus lift (x) characteristic as,

$$\Delta Cv = (\delta Cv/\delta x).\Delta X$$ ... Equation (3)

The corresponding instantaneous change in the flowrate being controlled can be estimated from standard ISA equations. This can be used directly in the case of flowrate control, or indirectly to quantify controllability. Alternately, the procedure may be applied in the reverse direction, starting from desired controllability and then determining positioning accuracy, ΔX, desired.

The higher pneumatic stiffness of double-acting piston actuators easily allows positioning accuracy of 0.5%, where required; in comparison, positioning accuracy of 2 to 4% is typical for diaphragm type actuators.

Hysterisis, deadband, and and control stability requirements are all quantifiable based on the process being controlled. In critical applications such as the CVCS level control valves in PWR s and D2O valves in PHWR s, such measures are useful in terms of problem-solving as well as in evaluating options. In both cases, the high gain in the process loop forces stringent requirements in valve performance.

The last feature in operability is good control under transient conditions. This is a function of dynamics that includes actuator characteristics, valve internal characteristics and the process. The first two must match up with what the process control requires. In particular, the impact of subsystem design on valve requirements needs to be quantified where it is significant.

Environmental Qualification

Environmental qualification for safe use in nuclear environment is yet another aspect beyond this.

As a generic issue, safety-related valves have defined functions that the valve must be capable of performing prior, during and/or after an accident. While the experience of many years of operations has led to better definition of qualification requirements to assure such levels of reliability, there are still some gray areas. One specific case is that of "OFF" mode versus "ON" mode qualification. In most cases, it is assumed that the valves are fail-closed or fail-open, depending on the requirements; further, it is assumed a mechanical spring in the actuator for fail-action will take the valve to the desired position when the power is turned off, even if the diaphragm and seals were to fail - this may be termed as OFF mode qualification.

But what if the application requires power and the seals to operate in case of an accident ? Such a requirement would require more rigorous qualification and has to be evaluated for each such application. This may be termed as ON mode qualification and may be required in some applications.

Conclusion

In conclusion, severe service applications must be treated differently from general service control valves. In addition, different severe service applications have their own specific set of requirements for high performance. The performance limits of a control valve in such services is clearly a function of the technologies it features.

In order to avoid control valve problems in severe service, it is important to evaluate if the technology in it is suitable for the specific application.

Generic guidelines, which include quantitative and verifiable design rules, that address the root cause of problems in severe service applications are available. Adherance to these rules narrows the selection of technologies to those that would result in good long-term term performance, high reliability and eliminate the MWe losses associated with these valve applications.

Finally, incorporation of specific design rules in the valve specification provides objective and verifiable criteria that can be used to qualify the technology for good valve performance in service.

References

1. Persad, J., Scurr, M.J., Alikhani S., and Miller, H.L., Condenser Dump Valve Retrofits Solve Vibration Problems , Sixth EPRI Valve Technology Symposium, Portland, Maine, 1997

2. Miller, H.L., and Stratton, L., Fluid Kinetic Energy As A Selection Criteria For Control Valves , ASME Paper FEDSM97-3464

3. Leimkuehler, B., and Sherikar, S.V., Getting Optimum Performance Through Feedwater Control Valve Modifications , , Sixth EPRI Valve Technology Symposium, Portland, Maine, 1997

4. Ritz, G., Control Valve Actuator Options , *Control*, June 1994.

Appendix I : Recommended Specifications for Severe Service Control Valves

Liquid Applications

1.0 Trim Exit Velocity (see Figure 1 for location of trim exit velocity measurement)

1.1 For liquids, the Trim Exit Velocity shall not exceed 100 ft/sec. (30 m/sec.).

1.2 For flashing service, the Trim Exit Velocity shall not exceed 75 ft/sec. (22.5 m/sec.).

1.3 Supplier shall provide calculation demonstrating satisfactory compliance to the above mentioned Trim Exit Velocity requirements.

2.0 Pressure Reducing Stages

2.1 Supplier shall provide a sufficient number of discrete pressure drop stages to insure the elimination of:

- Vibration
- Erosive Action
- Cavitation

2.2 Supplier shall identify the number of pressure drop stages in proposed equipment.

3.0 Flow Direction

3.1 Flow direction shall be a flow to close (over-the-plug) configuration.

4.0 Protection of Valve Internals

4.1 Suppliers shall provide means to protect the valve internals from foreign particles such as weld slag.

Gas and Steam Applications

1.0 Velocity Head

1.1 Velocity Head in the trim shall be less than 70 psia (480 kPa) in order to eliminate:

- Vibration
- Erosive Action

Velocity Head is defined as:

$$V_h = \frac{\text{Density} \times V^2}{2g_c}$$

where gc is gravitational constant and V is the trim exit velocity.

1.2 Supplier shall provide calculation demonstrating meeting the above Velocity Head requirements.

2.0 Pressure Reducing Stages

2.1 Supplier shall provide a sufficient number of discrete pressure drop stages to insure the elimination of:

- Vibration
- Erosive Action

2.2 Supplier shall identify the number of pressure drop stages in proposed equipment.

Liquid, Gas and Steam Applications

1.0 Noise

1.1 The maximum allowable noise level shall be 85 dBA or less at 3 feet (1 meter) from the downstream bare pipe surface.

1.2 The specified noise level shall be attained without the use of orifices, mufflers, diffusers, and/or credit for thermal or acoustic insulation.

1.3 Supplier shall provide calculation demonstrating meeting the above noise requirements.

2.0 Valve Trim

2.1 Valve shall have quick-change type trim utilizing top entry. No components shall be screwed or welded into the body.

2.2 The valve shall have equal pressure distribution around the plug.

3.0 Seating Forces

Based on the specific leakage class, the valve shall have as a minimum the seating forces shown below:

3.1 Class IV (FCI70-2) - 400 lb. per linear inch of seat ring circumference (7 kg/mm).

3.2 Class V (FCI70-2) - 700 lb. per linear inch of seat ring circumference (13 kg/mm).

3.3 MSS - SP61 (Block Valve)

For less than 3,000 psi (21 Mpa) 1,000 lb. per linear inch of seat ring circumference (18 kg/mm).

For greater than 3,000 psi (21 Mpa) 1,500 lb. per linear inch of seat ring circumference (27 kg/mm).

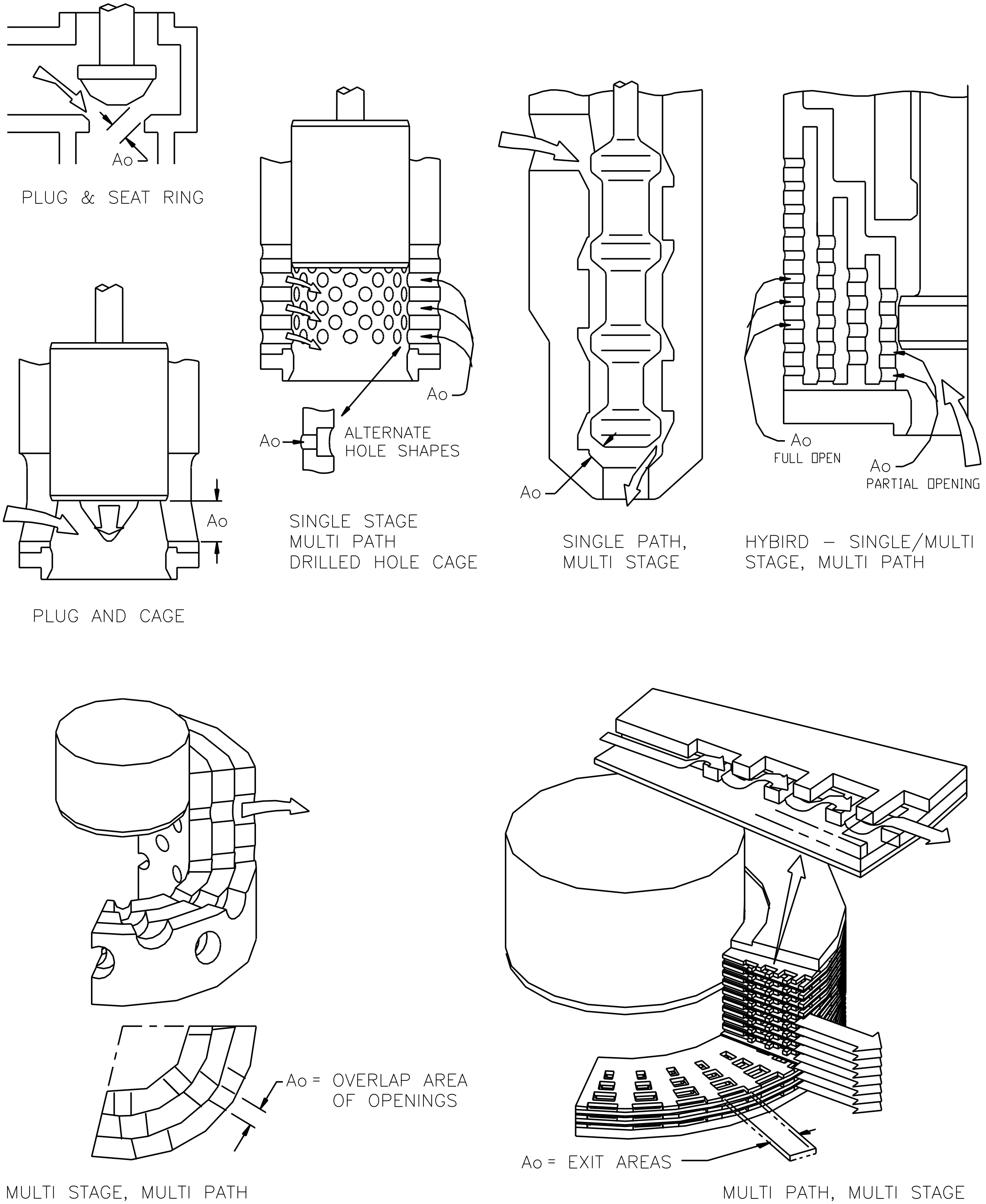

Figure 1
Different approaches for controlling fluid kinetic energy in control valves
(Reference 1)

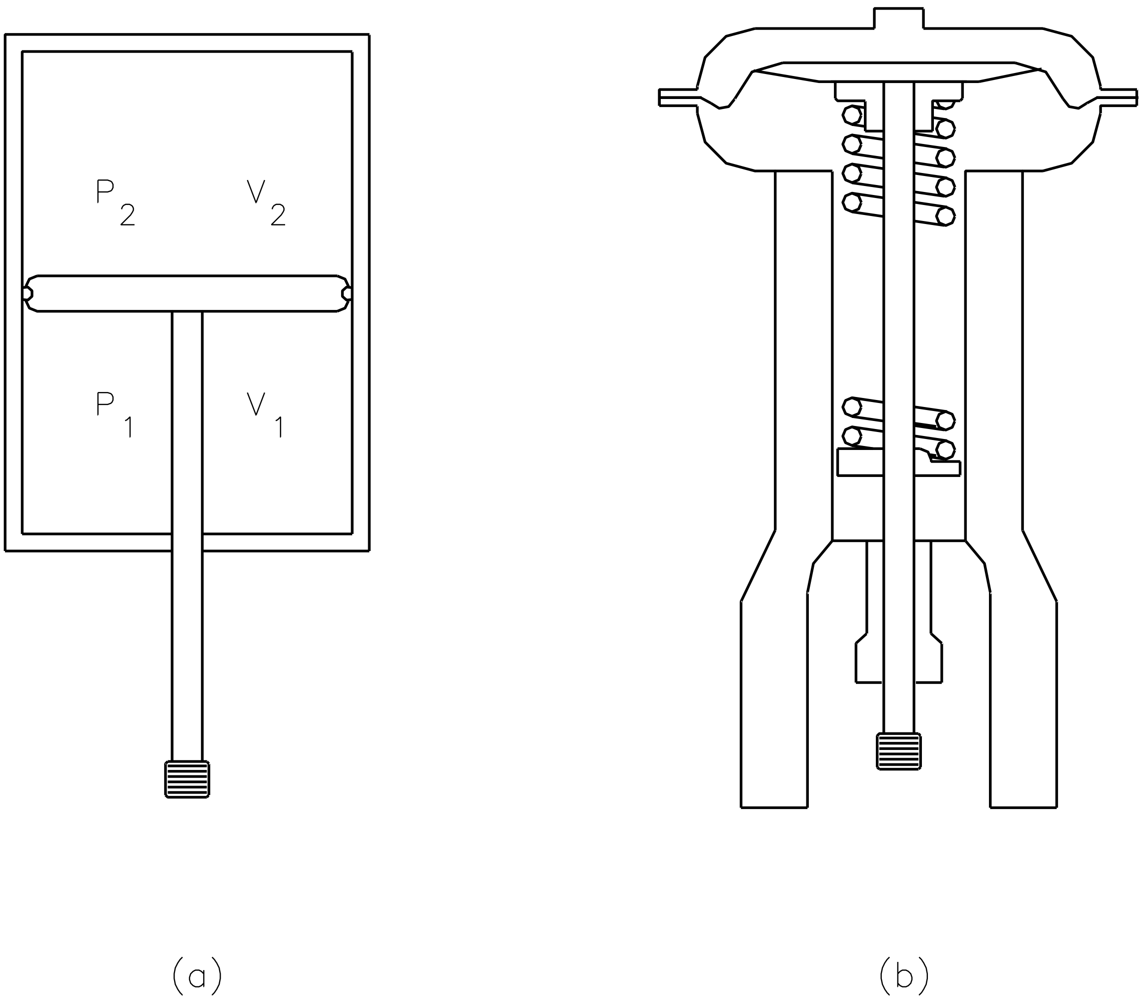

Figure 2

Nomenclature for calculation pneumatic stiffness of actuators:
(a) Pneumatic double acting piston type, and
(b) Diaphragm type.
A = Piston or diaphragm area
P_1, P_2 = pressure, psia
V_1, V_2 = Volume of air, cu-in
L_1, L_2 = equivalent air - volume lengths
$L_1 = V_1/A$
$L_2 = V_2/A$

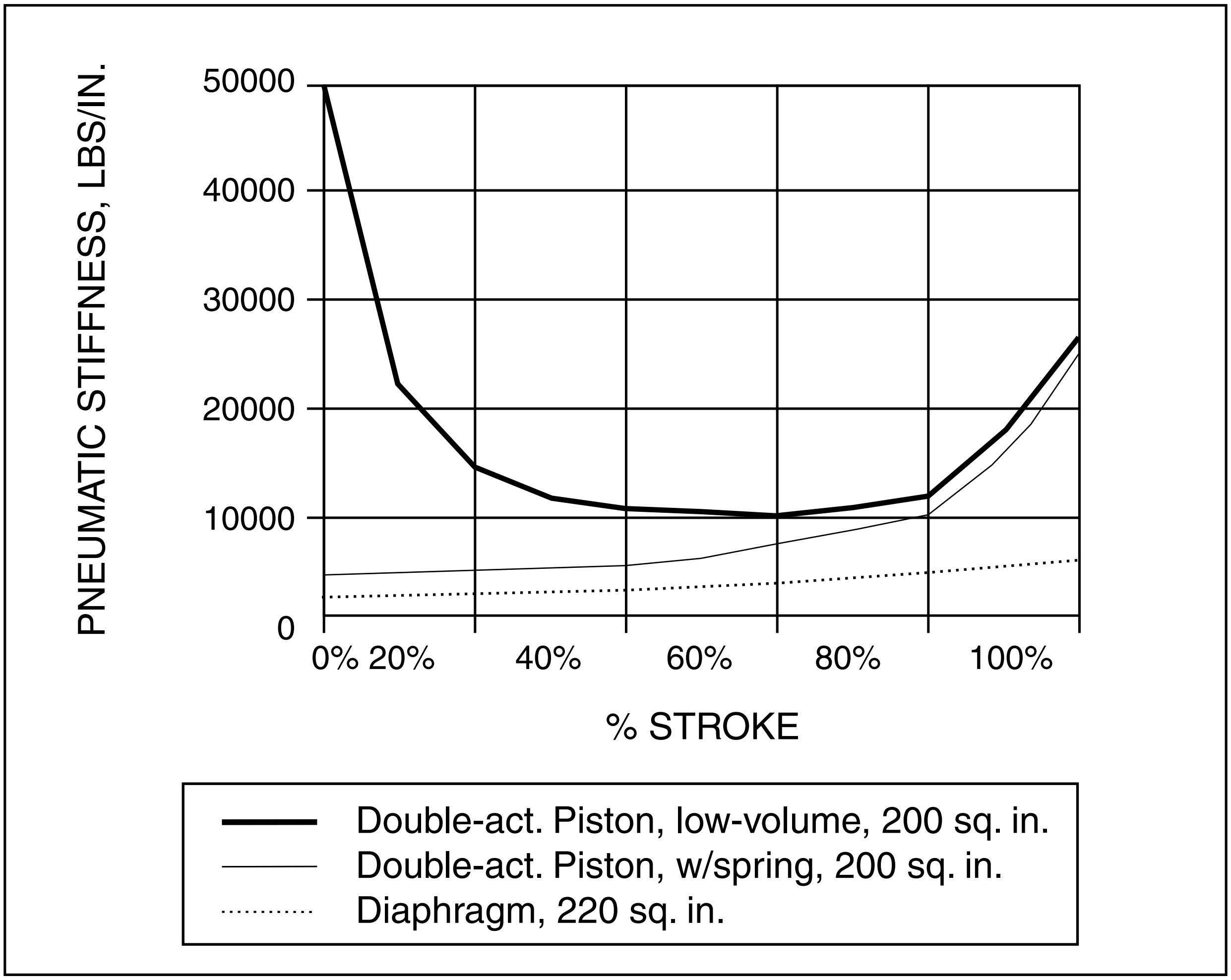

Figure 3

Comparison of pneumatic stiffness of (a) 200 sq. in. low-volume double acting (D.A.) piston actuator, (b) 200 sq. in. double-acting piston actuator with spring, and, (c) 220 sq. in. diaphragm actuator.